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СУДОВЫЕ ВСПОМОГАТЕЛЬНЫЕ МЕХАНИЗМЫ И СИСТЕМЫ

ИЗДАТЕЛЬСТВО МОРСКОЙ ТРАНСПОРТ
МОСКВА

На английском языке

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MARINE AUXILIARY MACHINERY and SYSTEMS

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PREFACE

The textbook "Marine Auxiliary Machinery and Systems" consists of three parts: Marine Fluid-Pressure Mechanisms, Specialized Auxiliary Machinery and Shipboard Systems.

Each part deals with the principles and the fundamentals of the theory, design and operation of a definite group of auxiliary machinery.

The theory involved is presented on a level which enables the student to understand the operation of the machines, to evaluate their operational characteristics, to make verifying calculations and select machinery for specific cases, and also to properly analyze new designs and master their operation. The author regards the theoretical, design and reference material presented in the textbook as the basis for carrying out normal operation and maintenance of auxiliary machinery, but not as a source providing information on their design or complete calculation.

The book does not deal with the auxiliary equipment of marine plants which is an integral part of these plants and depends upon their type, being, therefore, included in the corresponding courses of study.

The first part of the textbook is concerned with reciprocating, rotary, centrifugal, propeller and jet pumps, fans and fluid drives.

The second part considers only specialized auxiliary machinery.

The third part contains all necessary information on shipboard systems and the special systems of ice-breakers and tankers.

INTRODUCTION

Machinery designed for providing normal operation of the shipboard systems, gear and installations and constituting their integral part is called auxiliary machinery.

Auxiliary machinery maintains the required operating and manoeuvring properties of the ship both at sea or in harbour, and ensures its safety. Auxiliary machinery is classified as fluid-pressure mechanisms and deck machinery.

Fluid-pressure mechanisms include the pumps and fans of the shipboard systems and marine power plants, as well as fluid drives.

Deck machinery includes the mechanisms of the steering arrangement and of the hoisting gear.

Hoisting gear includes anchoring, warping, cargo-handling, towing, boat-handling and ash-disposal gear.

In accordance with their functions, shipboard gear and deck machinery are classified as mechanisms of the:

(1) steering gear, called steering engines,

(2) anchoring and warping arrangements, called windlasses and anchoring and warping capstans,

(3) warping, or mooring, appliances, called warping capstans and winches,

(4) cargo-handling equipment, called cargo winches,

(5) boat-handling gear, called boat winches,

(6) ash-disposal equipment, called ash hoisting winches,

(7) towing equipment, called towing winches.

The rigid demands imposed by shipboard conditions on the effective utilization of floor space and cubic capacity of engine and boiler rooms compel us to employ the same mechanism for maintaining the operation of two and sometimes more systems and arrangements. Thus, for example, the ballast system pumps may be employed for the needs of the ballast, drainage, fire, sanitary and ash-disposal systems.

Another example of using one mechanism for two purposes is the windlass which actuates both the anchoring and warping appliances. Such a combination of functions, however, should ensure,

if necessary, simultaneous operation of the respective systems and arrangements without in any way impairing their effectiveness.

According to their purpose, standby, or emergency, auxiliary machinery is provided on the ship to ensure normal operation and manoeuvrability when the main auxiliary machinery is out of order.

The effectiveness of auxiliary machinery is based on:

(1) its capability of ensuring normal operation of the respective systems or arrangements under any operating or navigating conditions,

(2) its low weight and small overall size,

(3) its high economy,

(4) its low cost,

(5) its convenient operation and repairs,

(6) its dependability,

(7) its ability to operate for long periods without repairs.

The capability of ensuring normal operation under any possible operating and navigating conditions is achieved by meeting the corresponding requirements for the design, manufacture, assembly and installation of the auxiliary machinery.* Auxiliary machinery will have a low weight and a small overall size if high-quality materials are used in its manufacture, if more exact formulas are used in its design, if the units are more rationally arranged and if their speed is increased. Regardless of its purpose any auxiliary machine consists of a motive unit and operational unit or units.

Therefore, economical operation of a particular auxiliary machine depends upon the efficiencies of both the motive unit and the operational units.

Economical operation of electric- and steam-driven auxiliary machinery can be appraised by the hourly fuel consumption per horse power developed by the motive unit of the machine in operation.

Depending upon its purpose and type of ship on which it is installed, the motive units of auxiliary machinery may be steam engines, steam turbines, electric motors, hydraulic motors and internal combustion engines.

Previously, steam engines were the main types of motive units of ships having steam power plants, and electric motors—of motorships. Present-day steamships frequently have auxiliary machinery powered by electric motors which are supplied with energy produced by economical generators with economical steam drives.

At the same time, motorships are being equipped with auxiliary machinery powered by steam engines which are supplied with

* Many of these requirements are stipulated by the U.S.S.R. Shipping Register.

steam produced in waste-heat boilers heated by the exhaust gases of the ship's internal combustion engines when the ship is under way, and by liquid fuel in harbour.

Hydraulic motors are used to some extent to power the auxiliary machinery of passenger ships because of their silent operation and infinitely variable shaft speeds.

Internal combustion engines drive auxiliary machinery on ships which do not have a power plant. They are also used to power standby or emergency machinery.

A specific feature of the motive units of auxiliary machinery is their low power rating, not exceeding several dozen horse power even for the most powerful pieces of machinery. This is the reason for the comparatively low efficiency of this machinery.

The most economical and convenient in operation is auxiliary machinery powered by steam engines or electric motors. The latter, due to their exceptional convenience, especially in northern latitudes, are finding ever-increasing application.

Due to the relatively low efficiency of the motive units of auxiliary machinery, economical operation of an auxiliary installation will depend to a considerable extent upon properly determining the required power of the motive unit, rational selection of the operational unit and its efficiency and also upon a strict observance of the operating rules established for the machinery.

The fuel consumption of auxiliary machinery amounts to:

(1) 2.2 to 3.7 kg of coal per hp-h when operating on live steam from the boilers* or 1.5 to 2.7 kg of motor fuel per hp-h;

(2) 1.74 to 6 kg of motor fuel per hp-h when operating on steam from a waste-heat boiler (this fuel consumption is only conditional since waste heat is utilized);

(3) 1.42 to 1.86 kg of coal per hp-h or 0.96 to 1.32 kg of motor fuel per hp-h when operating on electric power produced by a steam dynamo;

(4) 0.22 to 0.243 kg of motor fuel per hp-h when operating on electric power produced by a diesel dynamo.

The fuel consumption of the auxiliary machinery of systems, gear and the main power installations usually constitutes 10 to 15 per cent of the fuel consumption of the main steam engines, but may reach 20 to 30 per cent on certain ships.

Increased fuel consumption may also be due to nonobservance of rules for the operation and maintenance of auxiliary machinery.

Attempts should be made to diminish the adverse effect of the auxiliary machinery on the economic efficiency of the steam power installations as a whole by substituting exhaust steam from this

* Refers to steam engines with a crank mechanism.

machinery for live steam in various units of the power installation, and by employing heating systems with steam take-off and with the utilization of spent steam from the auxiliary machinery for shipboard needs.

The cost of auxiliary machinery depends to a considerable extent upon the complexity of manufacturing and installation of the machinery, and also upon its weight.

The cost can be substantially reduced by unifying and standardizing this machinery. This also simplifies servicing, repairs and the making of spare parts. The problem of selecting the type of auxiliary machinery for a given ship and the kind of energy with which it is to be powered can be properly solved only after a comprehensive analysis of various alternative thermal systems of the whole power installation of the ship.

The type of auxiliary machinery used on ships varies with the operating and manoeuvring requirements specified for the ships.

Part One

MARINE FLUID-PRESSURE MECHANISMS

Chapter 1

PUMPS

A. MARINE PUMPS: GENERAL CONSIDERATIONS

1-1. Energy Equation of Liquid Flow and Principles of Pump Action

A stream of liquid under steady pipe flow conditions possesses definite amounts of potential and kinetic energy.

In the total flow of liquid let us consider a unit volume that weighs 1 kg and passes through the cross section *I* (Fig. 1*a*). This volume of liquid, located at a height of z_1 m above the datum, is subjected to a pressure of p_1 kg per sq m and moves at a velocity of v_1 m per sec.

The total specific energy of the kilogram of liquid considered at section *I*, as expressed by Bernoulli's equation, will be

$$E_1 = z_1 + \frac{p_1}{\gamma} + \frac{v_1^2}{2g} \text{ kg-m per kg} \qquad (1)$$

where z_1 = potential energy due to position, kg-m per kg

$\frac{p_1}{\gamma}$ = potential energy of hydromechanical pressure, kg-m per kg

$\frac{v_1^2}{2g}$ = kinetic energy, kg-m per kg

γ = density of the moving liquid, kg per cu m.

When the considered kilogram of liquid reaches cross section *II*, it will be at a distance of z_2 m from the datum and subjected to a pressure of p_2 kg per sq m. Here it will move with a velocity of v_2 m per sec.

The specific energy of 1 kg of liquid at cross section *II* will be

$$E_2 = z_2 + \frac{p_2}{\gamma} + \frac{v_2^2}{2g} \text{ kg-m per kg} \quad (2)$$

If we denote by ΔE the difference in energy inherent in the kilogram of liquid at the two cross sections, then

$$\Delta E = E_1 - E_2 = \left(z_1 + \frac{p_1}{\gamma} + \frac{v_1^2}{2g}\right) - \left(z_2 + \frac{p_2}{\gamma} + \frac{v_2^2}{2g}\right) \text{ kg-m per kg} \quad (3)$$

Therefore, if

$$E_1 = E_2 \text{ then } \Delta E = 0$$
$$E_1 > E_2 \text{ then } \Delta E > 0$$
$$E_1 < E_2 \text{ then } \Delta E < 0$$

The first case ($\Delta E = 0$) concerns the movement of a nonviscous liquid which loses no energy and receives none from any external source. Changes and conversion of one form of energy into another are possible in this form of flow but the total specific energy of the liquid remains constant.

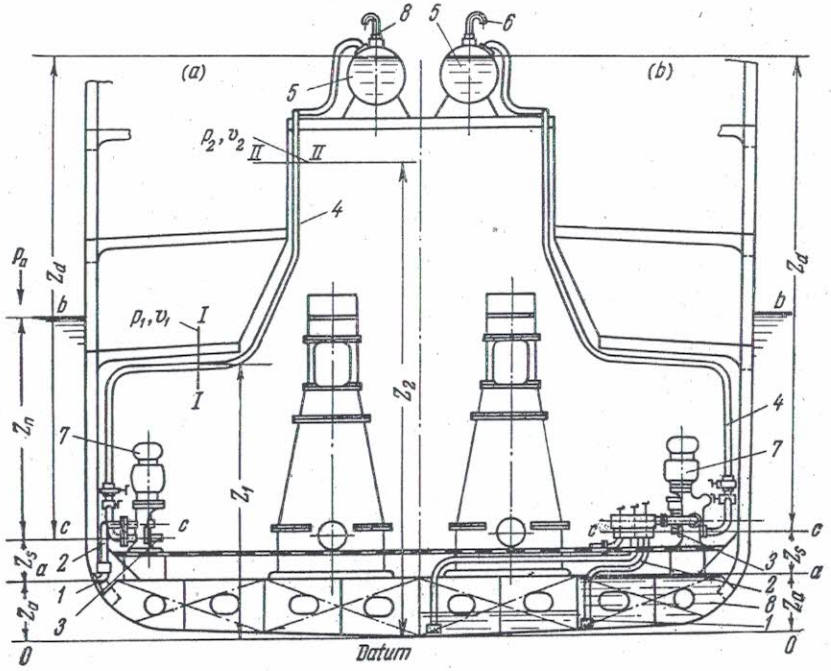


Fig. 1.

The second case ($\Delta E > 0$) is characteristic of viscous liquid flow. Consequently, part of the energy is expended in overcoming the hydraulic resistance encountered in flowing from cross section *I* to cross section *II*. Besides this, the inequality $\Delta E > 0$ will also be valid for the case in which energy is used between the sections *I* and *II* to drive a hydraulic motor performing useful work.

In the third case ($\Delta E < 0$) additional energy is transferred to each kilogram of liquid from an external source as it flows from section *I* to section *II* so that the specific energy at section *II* will exceed that at section *I*. This additional energy is transferred to the liquid as it flows through a pump and may be in the potential, velocity, or in both forms. Thus, the pump imparts energy to the liquid that is required to overcome resistance in its flow to its destination. The principle of operation of the pump depends upon the form of energy imparted to the liquid.

Any pumping plant consists of a pump, piping system, valves, tanks and instrumentation.

Energy imparted to the liquid by the pump is expended in overcoming resistance to flow in the piping and in passing through the valves, in changing the direction and velocity of flow, in lifting the liquid to a higher level, in obtaining a jet of the desired length and in overcoming the difference in pressure in tanks connected by the piping of the pumping plant.

The terms of Bernoulli's equation, expressed in metres of water column, represent the corresponding heads, namely:

z=elevation head, mH_2O (metres of water column)

$\frac{p}{\gamma}$=pressure head, mH_2O

$\frac{v^2}{2g}$=velocity head, mH_2O.

The sum of the members $z+\frac{p}{\gamma}$ constitutes the static head of the flow of liquid while $\frac{v^2}{2g}$ is its dynamic head.

Therefore, the total hydrodynamic head at any cross section of the pipeline will be

$$H = z + \frac{p}{\gamma} + \frac{v^2}{2g} \text{ m}$$

The difference in the total hydrodynamic heads in two cross sections of the flow, located before or after the pump or turbine, determines the loss of head h_l in the movement of the liquid from one section to the other, i.e.,

$$(z_1 - z_2) + \left(\frac{p_1}{\gamma} - \frac{p_2}{\gamma}\right) + \frac{v_1^2 - v_2^2}{2g} = h_l \text{ m}$$

The pumping plant may be designed for picking up outboard liquids, overboard discharge, transferring liquids along piping from tank to tank on board, supplying liquids to mechanisms, boilers, fire hoses, etc.

The potential energy imparted to the flow by the pump is expended in overcoming hydraulic resistance in the pipelines, in elevating the liquid and in overcoming the pressure at the points of delivery. The kinetic energy of the flow determines the velocity of the liquid in the piping and the length of the stream spouting from the point of discharge.

The magnitude of each item of energy expenditure depends upon the purpose and operating conditions of the shipboard pumping plant. Thus, for instance, in ballast, drainage and sanitary pumping arrangements the energy of the liquid is expended chiefly in overcoming hydraulic pipe resistance and in raising the liquid to a higher level, in boiler feed pumps it is employed to overcome the pressure in the boiler, while in fire-fighting equipment the energy mainly elevates the liquid and provides for a stream of the required length thrown by the fire nozzle.

Two principal types of pump arrangements in reference to the level of the liquid being pumped are employed in shipboard plants. The pump may be located either below (Fig. 1*a*) or above (Fig. 1*b*) the liquid level. Let us consider the principles of pump operation in the fresh and sea water pump systems shown in Fig. 1.

Due to the difference in water pressure inside and outside the suction pipes supplying pumps *3*, water passes through strainers *1* and suction pipes *2* to the pump which discharges the water through the delivery pipes *4* to the service tanks *5*, furnished with atmospheric pipes *6*. The latter maintain atmospheric pressure in the tanks. The main part of the energy supplied to pump motors *7* is imparted to the water by the pumps. This energy is required to elevate the water to a height of z_s+z_d m and to overcome the resistance of the water flowing through suction *2* and delivery *4* piping.

Assuming steady flow in the piping, the pressure at any cross section can be determined by Bernoulli's equation.

After selecting a datum and applying Bernoulli's equation to the free surface of the sea *b-b* and the pipe cross section at plane *c-c*, we can write the following equation for a plant located below the level of the fluid being pumped (Fig. 1*a*):

$$(z_n+z_s+z_a)+\frac{p_b}{\gamma}+\frac{v_b^2}{2g}=(z_s+z_a)+\frac{p_s}{\gamma}+\frac{v_s^2}{2g}+h_l \qquad (4)$$

where z_n = distance from sea level to plane *c-c* passing through the pump, m

p_b = sea-level pressure, kg per sq m
v_b = velocity of the liquid at sea level in the direction of the inlet orifice of the suction pipe, m per sec
z_s = distance of cross section *c-c* from free level *a-a* of the water in bilge tank *8*, m
z_a = distance from plane *a-a* to the datum, m
p_s = suction pressure at cross section *c-c* of the piping, kg per sq m
v_s = velocity of the liquid at cross section *c-c* of the piping, m per sec
h_l = loss of head in the suction piping within the section being considered, mH_2O.

Since the sea-level pressure p_b equals the atmospheric pressure p_a, this substitution can be made in equation (4), which then becomes

$$z_n+\frac{p_a}{\gamma}=\frac{p_s}{\gamma}+\frac{v_s^2}{2g}\left(1-\frac{v_b^2}{v_s^2}\right)+h_l$$

Since v_s is incomparably greater than v_b, we can write

$$\frac{p_s}{\gamma}=\frac{p_a}{\gamma}+z_n-\left(\frac{v_s^2}{2g}+h_l\right) \text{ m} \qquad (5)$$

If the effect of the vapour pressure head $\frac{p_v}{\gamma}$, corresponding to the temperature of the liquid being pumped, on the suction head is taken into consideration (Table 2) then

$$\frac{p_s}{\gamma}=\frac{p_a}{\gamma}+z_n-\left(\frac{p_v}{\gamma}+\frac{v_s^2}{2g}+h_l\right) \text{ m} \qquad (6)$$

It is necessary, for normal operation of the pump, that $\frac{p_s}{\gamma}$ be positive in value. This is true if

$$\frac{p_a}{\gamma}+z_n>\left(\frac{p_v}{\gamma}+\frac{v_s^2}{2g}+h_l\right) \text{ m} \qquad (7)$$

It is evident that normal operation of the given pumping plant depends upon the values of $\frac{p_a}{\gamma}$ and $\frac{p_v}{\gamma}$. The value of $\frac{p_a}{\gamma}$ varies, though only slightly, with the elevation of the pumping plant above sea level (Table 1).

The vapour pressure p_v of the liquid being pumped depends upon the temperature of the liquid and, at high temperatures, may reach a value at which normal pump operation is impaired. The

data of Table 2 show a substantial increase in vapour pressure when the temperature of the water is raised.

Table 1

Elevation above sea level, m		0	100	200	300	400	500	600	700	800	900	1000	1200	1500	2000
Atmospheric pressure	mmHg	760	751	742	733	724	716	707	699	690	682	674	658	635	598
	mH_2O	10.3	10.2	10.1	9.9	9.8	9.7	9.6	9.5	9.4	9.3	9.2	8.9	8.6	8.1

Table 2

Temperature of water being pumped, °C		0	10	20	30	40
Water vapour pressure	mmHg	4.5	9.16	17.39	31.55	54.91
	mH_2O	0.06	0.12	0.24	0.43	0.75

Temperature of water being pumped, °C		50	60	70	80	90	100
Water vapour pressure	mmHg	91.98	148.79	233.08	354.52	525.39	760
	mH_2O	1.25	2.02	3.17	4.82	7.14	10.33

If necessary, the inequality (7) can be achieved by increasing the height z_n which is called the negative lift. Under specified conditions, z_n can be determined from equation (7):

$$z_n > \left(\frac{p_v}{\gamma} + \frac{v_s^2}{2g} + h_l\right) - \frac{p_a}{\gamma} \text{ m}$$

Considerable difficulty is encountered in accurate determination of the vapour pressure at various temperatures for petroleum products, since the latter are mixtures of various hydrocarbons that boil at different temperatures. In such cases, the vapour pressure is determined on the basis of the hydrocarbons with the lowest boiling points. Approximate vapour pressure values, sufficiently accurate for operational calculations, can be found in Fig. 2.

Pumping installations with a positive pressure at the suction end are called negative-lift pumping plants.

Pumping installations that handle hot liquids or liquids with a viscosity in excess of 8 or 10 °E (Engler degrees) operate with negative lift as do many shipboard pumps handling cold liquids due to the specific features of their installation.

We can now pass over to the equation for the suction head of a pump installation located above the level of the liquid being pumped (Fig. 1*b*). Bernoulli's equation takes the following form when ap-

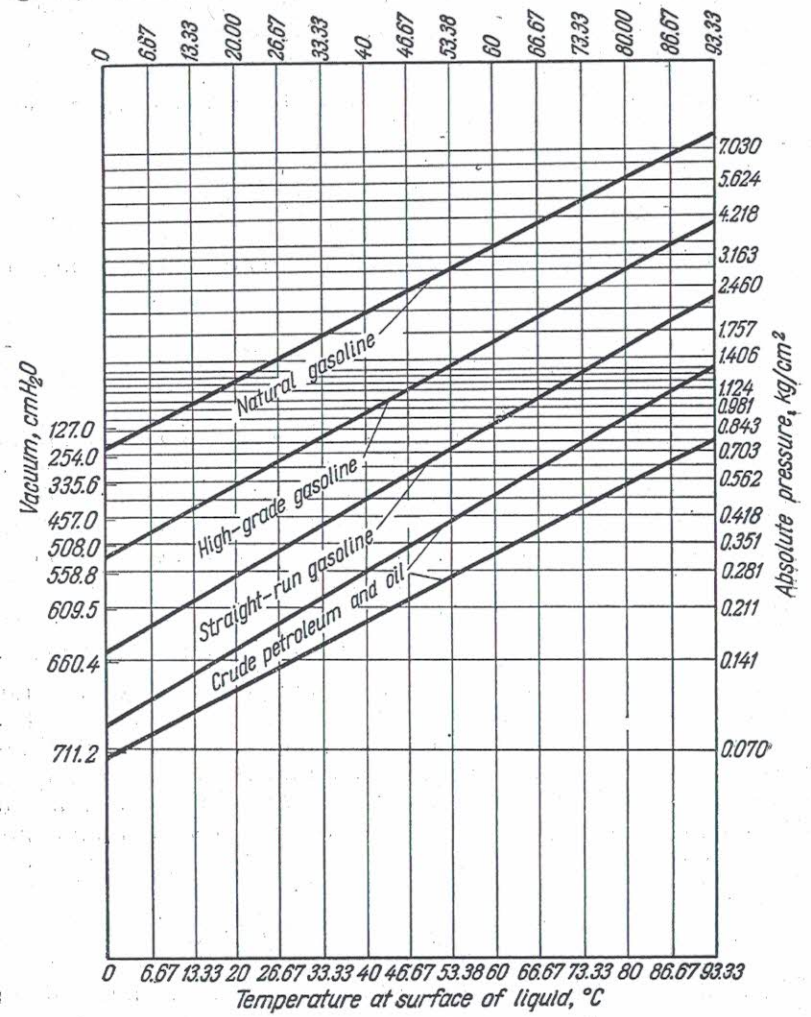


Fig. 2.

plied to the liquid surface *a-a* in bilge tank *8* and plane *c-c* passing through the pump:

$$z_a + \frac{p_a}{\gamma} + \frac{v_a^2}{2g} = (z_a + z_s) + \frac{p_s}{\gamma} + \frac{v_s^2}{2g} + h_l \qquad (8)$$

where $\frac{p_a}{\gamma}$ = pressure head on the free surface of the liquid in bilge tank 8, equal to the atmospheric pressure, mH_2O

v_a = velocity of the liquid on the free surface a-a in the direction of the inlet screen, m per sec

v_s = velocity of the liquid at cross section c-c, m per sec.

Solving equation (8) for $\frac{p_a}{\gamma}$ and rearranging the terms, we obtain

$$\frac{p_a}{\gamma}=z_s+\frac{p_s}{\gamma}+\frac{v_s^2}{2g}\left(1-\frac{v_a^2}{v_s^2}\right)+h_l$$

Since v_s is incomparably greater in value than v_a, the suction pressure head equals

$$\frac{p_s}{\gamma}=\frac{p_a}{\gamma}-\left(z_s+\frac{v_s^2}{2g}+h_l\right) \text{ m} \qquad (9)$$

It is evident from equation (9) that the pressure head in the suction line is less than the atmospheric pressure. This indicates the presence of a partial vacuum which enables the liquid to enter the pump due to unbalanced atmospheric pressure on the free surface of the liquid in the bilge tank. Thus, for normal operation of a pump located above the liquid being pumped, it is necessary that the pump create a partial vacuum in the suction pipe and that the pressure head $\frac{p_s}{\gamma}$ be positive in value along the full length of the suction line. Due to the fact that the suction pressure in a pumping plant with lift is always lower than atmospheric pressure, vapour may be evolved from the liquid being pumped. The liquid begins to boil when the pressure p_s in the suction pipe reaches the value p_v. This leads to rupture of the flow of liquid in the suction pipe and failure in pump lift. Therefore, normal operation of a pumping installation requires that $p_s > p_v$, or

$$\frac{p_s}{\gamma}\geqslant\frac{p_a}{\gamma}-\left(\frac{p_v}{\gamma}+z_s+\frac{v_s^2}{2g}+h_l\right) \text{ m}$$

The maximum theoretical height (lift) $z_{s\,max}$ to which a liquid can be raised in the suction pipe is found if $\frac{p_v}{\gamma}$ is substituted for $\frac{p_s}{\gamma}$ and $z_{s\,max}$ for z_s in equation (9). Then

$$\frac{p_v}{\gamma}=\frac{p_a}{\gamma}-\left(z_{s\,max}+\frac{v_s^2}{2g}+h_l\right)$$

and, consequently,

$$z_{s\,max}=\frac{p_a}{\gamma}-\left(\frac{p_v}{\gamma}+\frac{v_s^2}{2g}+h_l\right) \text{ m} \qquad (10)$$

Since the atmospheric pressure p_a = 10,330 kg per sq m and $\frac{p_a}{\gamma}$ = 10.33 mH_2O, the lift for water in the suction pipe, even in the ideal case, in the absence of all losses, cannot exceed 10.33 m. The actual lift is considerably lower than the theoretical value.

It is obvious from equations (5) and (9) that a pumping plant located below the level of the liquid being handled operates under more favourable conditions, from the point of view of the suction pressure, than a pump operating with positive lift. Other conditions being equal, the suction head of a negative-lift installation is greater than for one operating with lift by the value of the head z_n+z_a, produced by a column of liquid of a height equal to the distance from the centre of the inlet orifice in the suction pipe to the free surface of the liquid being handled.

1-2. Principal Ratings of Pumps

The principal ratings of pumps are the output, head, suction lift, power and efficiency.

The output, or capacity, can be expressed either in volumetric or in weight units.

The volumetric capacity Q of a pump is defined as the volume of liquid delivered by the pump into the discharge pipe in unit time. It is measured in cu m per hour, cu m per sec, litres per sec, etc.

The weight capacity G of a pump is defined as the amount of liquid by weight delivered by the pump into the discharge pipe in unit time and may be expressed in tons per hour, tons per min, tons per sec, kg per hour, kg per min or kg per sec.

The theoretical capacity, or displacement, Q_t or G_t, is distinguished from the actual capacity, Q or G.

The ratio

$$\frac{Q}{Q_t}=\frac{G}{G_t}=\eta_v$$

is called the volumetric efficiency of a pump and determines the reduction in actual output as compared to the theoretical possible value.

The head (pressure) created by a pump is defined as the increment of energy imparted to each kilogram of liquid passing through the pump. It is expressed in mH_2O (or kg per sq cm).

The suction lift, or simply lift, is the loss of head required to overcome resistance in the suction line of the pumping plant; it is measured in mH_2O.

The useful power of a pump is the energy increment in the flow of liquid passing through the pump in unit time and is expressed in horsepower or kilowatts. Thus,

$$N_u = \frac{GH}{60\times60\times75} \text{ hp} = \frac{GH}{60\times60\times102} \text{ kW}$$

$$N_u = \frac{Q\gamma H}{60\times60\times75} \text{ hp} = \frac{Q\gamma H}{60\times60\times102} \text{ kW}$$

where H = the actual head created by the pump, mH_2O.

The mechanical efficiency, η_m, of a pump determines the loss in energy in its operation and enables the required power input N to be calculated:

$$N = \frac{N_u}{\eta_m} \text{ hp (kW)}.$$

1-3. Pump Classification According to Purpose and Principle of Operation

(A) PUMP CLASSIFICATION ACCORDING TO PURPOSE

In accordance with their purpose, shipboard pumps can be divided into three groups:

1. General service pumps whose function is to ensure the seaworthiness of the ship and to provide for the domestic needs of the crew and passengers, and also to maintain the necessary sanitary conditions on board.
2. Pumps of the shipboard systems, designed to serve the main and auxiliary systems, and to facilitate the maintenance of normal conditions for their operation.
3. Special-purpose pumps in tankers, trawlers, ice-breakers, life-saving ships and dredgers.

General service pumps include:

(1) bilge pumps,
(2) sanitary pumps,
(3) fire pumps,
(4) emergency pumps.

Bilge Pumps

Bilge pumps include ballast and drainage pumps.

Ballast Pumps. The purpose of these pumps is to take in liquid ballast from overboard, to fill the ballast tanks and peaks, to empty them of ballast and to transfer the ballast from certain tanks and peaks to others.

The capacity of ballast pumps must be sufficient to enable the tanks they serve to be emptied within 4 to 10 hours, depending upon the size of the ship.

Tanks that hold from 20 to 265 cu m of ballast are usually pumped dry within 1 to 3.5 hours; those holding from 360 to 1300 cu m are emptied within 4 to 5.5 hours.

The required capacity of a ballast pump can be determined from the formula

$$Q_b = 0.2825 d_b^2 v_b \text{ cu m per hr} \qquad (11)$$

where d_b = diameter, in cm, of the inlet pipe of the largest ballast tank. According to the regulations of the U.S.S.R. Shipping Register this value is to be taken from Table 3, depending upon the tank capacity in tons

$v_b = 2$ to 2.5 = velocity of water flow in the input line of the pump, m per sec.

Table 3

Tank capacity, tons	Inside diameter of pipe and fittings, mm	Tank capacity, tons	Inside diameter of pipe and fittings, mm
Up to 20	60	265 to 360	125
20 to 40	70	360 to 480	140
40 to 75	80	480 to 620	150
75 to 120	90	620 to 800	160
120 to 190	100	800 to 1000	175
190 to 265	110	1000 to 1300	200

At a water velocity $v_b = 2$ m per sec (recommended by the U.S.S.R. Shipping Register) the required ballast pump capacity will be

$$Q_b = 0.565 d_b^2 \text{ cu m per hr.} \qquad (12)$$

Because of water leakages this calculated capacity must be increased by 5 or 10 per cent. Ballast pump capacities range from 60 to 300 cu m per hour. The number of ballast pumps is not stipulated by the regulations of the U.S.S.R. Shipping Register.

Any pump of suitable capacity in a shipboard installation, except drinking-water pumps, can be employed for ballasting operations if the ballast tanks are not used to store liquid fuel. In the latter case, the use of standby cooling pumps of the internal combustion engines and the fire pumps for ballasting duty is prohibited.

Self-contained ballast pumps must be installed on oil tankers to serve the fore ballast tank.

All ballast pumps must have self-contained drives.

Drainage Pumps. These pumps remove small amounts of water that collect in the bilge courses and wells of the ship's hull. The sources of this water are sweating of the plating and the penetration of outside water through impaired seams in the hull. Moreover, water may enter the hull through openings in the deck and freeboard (that are not completely watertight) during stormy weather, rain, deck hosing or washing the deck structures and holds.

Water enters the bilge courses and wells of the engine and boiler rooms and the propeller shaft tunnels through leaky joints in the piping and parts of the machinery, from the blow-off systems of the mechanisms, from systems for intermittent cooling of friction surfaces, from the ash-extinguishing arrangements in the boiler room, etc. Drainage pumps are classified into engine and boiler room drainage pumps and bilge drainage pumps, according to their functions.

In the great majority of cases, the same pump is employed for both duties. According to the Register, the flow velocity in the drainage piping should be at least $v_d=2$ m per sec for pipe diameters equal to:

$$d_d = 25 + \sqrt{2.8L(B+H)} \text{ mm} \qquad (13)$$

for the drainage main and branch lines leading directly to the pump, and

$$d_d = 25 + \sqrt{4.5l(B+H)} \text{ mm} \qquad (14)$$

for the inlet branches of compartments and tanks, and also when hand-operated pumps are installed,

where L=maximum length of the ship, m

B=breadth amidships, m

H=height of the side to the deck of the watertight bulkheads (according to the design drawings);

l=length of the compartment to be drained, m.

Regardless of the results obtained from formulas (13) and (14), the diameter of the inlet branches must not be less than 50 mm.

The capacity of a drainage pump is selected to suit the pipe diameter of the drainage main and should not be less than the value given in Table 4.

The Register stipulates a minimum of two drainage pumps with self-contained driving facilities on each ship. Any of the general service pumps of suitable capacity on the ship may serve as one of the drainage pumps.

On ships powered by reciprocating steam or internal combustion engines, one of the drainage pumps (for engine-room bilge duty) may be driven from the main engines.

Table 4

Inside diameter of the drainage main, mm	Capacity of each drainage pump, cu m per h	Inside diameter of the drainage main, mm	Capacity of each drainage pump, cu m per h
50	15	133	103
57	19	140	113
64	23	146	124
70	28	152	135
76	34	158	146
82	40	165	158
89	46	171	171
95	53	178	183
103	60	184	197
108	68	190	210
114	76	197	224
120	84	205	240
127	93		

All sea-going ships with a main engine rating over 300 hp must have an emergency drainage pump for the engine room. The condenser circulating pump on steam ships or the main engine cooling pump on motor ships may be employed for this purpose.

Hand-operated or water-jet pumps may be used for drainage duty on ships with a main engine rating up to 200 hp.

A drainage pump may be employed as a standby ballast and fire pump, but in the latter case, it must have no connections with compartments in which oil or liquid fuels are present. The total output of the drainage pumps should never be less than that of the fire pumps.

Sanitary Pumps

These pumps are known as water supply or fecal pumps, depending upon the liquid they handle.

Water supply pumps are employed to deliver water to the service tanks as well as to the air-pressure tanks from where it flows to the users. Water supply pumps may be either for sea water or fresh (washing and drinking) water.

Fecal pumps are used to empty the sewage tanks of sewage accumulated during berthing in a harbour or when navigating waters whose pollution by fecal liquids is prohibited. The required capacity of water pumps can be calculated on the basis of the following water consumption requirements per person:

sea water . . . . . 20 to 30 litres per day,
washing water . . . . . 20 litres per day,
drinking water . . . . 10 litres per day.

The pump capacity should be such that service tanks, holding from 0.2 to 0.4 of the daily consumption requirements, are filled within 1 to 6 hours, depending upon the type and size of the ship.

Water supply pumps of passenger ships have capacities ranging from 5 to 12 cu m per hour and higher, while those of cargo vessels and tugs have capacities from 3 to 5 cu m per hour.

The capacity of a fecal pump depends upon the design volume of the sewage tank and the duration of berthing in ports and navigation in waters where fecal discharge is prohibited. In any case, the capacity should be sufficient to discharge the sewage from the tank in 1 or 2 hours. The volume of the sewage tank can be calculated on the following per capita sewage values: 12 litres per day from toilet bowls, 2 litres per day from urinals and 2 to 5 litres per day from other sanitary equipment.

Fire Pumps

Fire pumps supply water to the fire-extinguishing appliances.

The capacity of the fire pumps must suffice to ensure simultaneous operation of:

(a) 15 per cent of all the fire risers installed on the ship, but not less than three,

(b) automatic sprinkler and drencher systems,

(c) the foam-making installation of the largest compartment on the ship or the adjacent communicating compartments,

(d) deck-flooding systems of oil tankers.

The head developed by each fire pump should be sufficient to produce a stream thrown from the orifice of the fire nozzle to a height of 12 m above the highest deck structure when all the arrangements mentioned above operate simultaneously.

The water thrown Q_s, in litres per sec, per nozzle of the fire-fighting system for a given nozzle orifice size d_n, in mm, and height of stream h_n, in m, can be determined from V. Lobachov's chart (Fig. 3, lines *ab* and *bc*).

The head required in the nozzle to throw the stream of water to the height h_n is found from formula

$$H_n = \frac{Q_s^2 10^4}{0.12106 d_n^4 \mu^2} \text{ mH}_2\text{O}$$

where $\mu \approx 1$ = consumption factor of the nozzle.

This formula can also be used to determine the head necessary to throw a stream through a distance of h_{ds} from a drencher or sprinkler. Here, $\mu \approx 0.7$ to 0.75.

The head H_h required at the fire hydrant to throw Q_s litres per sec of water in a stream to a height of h_n metres can be found on the chart at the intersection of the extension of line *bc* with a horizontal line

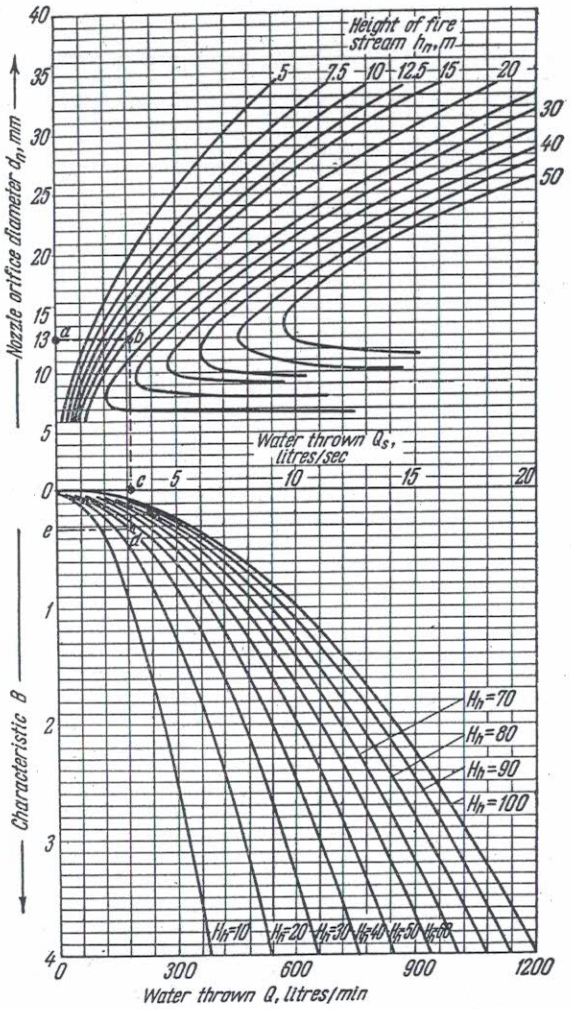


Fig. 3.

drawn from point *e*. The latter corresponds to the value of characteristic *B* which is taken from the table for the given size of hose. The values determined in the example in Fig. 3 are: Q_s=3.15 litres

Hose diameter d_h, mm	Hose length l_h, m	Nozzle orifice diameter d_n, mm: 10	13	16	19 — Characteristic B	Hose diameter d_h, mm	Hose length l_h, m	Nozzle orifice diameter d_n, mm: 10	13	16	19 — Characteristic B
50	0	0.121	0.346	0.793	1.577	65	0	0.121	0.346	0.793	1.577
	10	0.119	0.331	0.722	1.320		10	0.1205	0.342	0.776	1.51
	20	0.118	0.318	0.622	1.130		20	0.120	0.339	0.758	1.44
	40	0.114	0.304	0.568	0.882		40	0.1195	0.332	0.726	1.33
	60	0.111	0.274	0.498	0.723		60	0.1185	0.326	0.696	1.23
	80	0.108	0.257	0.442	0.612		80	0.118	0.320	0.669	1.15
	100	0.105	0.241	0.398	0.531		100	0.117	0.314	0.644	1.08

per sec, $B=0.318$ and $H_h=29$ m when $d_n=13$ mm, $h_n=20$ m and the hose diameter and length are $d_h=50$ mm and $l_h=20$ m, respectively. The amount of water thrown by one sprinkler or drencher can be found from the curves in Fig. 4.

The amount of water required for the operation of a foam extinguishing installation, covering an area of F_f sq m with foam in 10 minutes, is determined from the formula

$$Q_f = 0.6 F_f h_f \text{ cu m per hour} \qquad (15)$$

where F_f=area covered by foam, in sq m
h_f=thickness of the foam layer, in m.

The U.S.S.R. Shipping Register stipulates the following thicknesses of foam blanket produced by a chemical foam generator that are required for extinguishing: gasoline (petrol)—22.5 cm, kerosene—17.5 cm, and masut, heavy petroleum or oil—10 cm.

Foam blankets produced by compressed air foam-making systems must be of the following thickness: 50 cm for petrol, 30 cm for kerosene or naphtha, and 20 cm for masut, heavy petroleum or oil.

The water consumption of a deck-flooding system may be calculated by the formula

$$Q_{df} = \mu_{df} n_{df} \omega_{df} l_{df} \sqrt{2gH_{df}} \text{ cu m per sec} \qquad (16)$$

where $\mu_{df}=0.62$=consumption factor
n_{df}=number of holes per metre of the flooding pipeline
l_{df}=length of the flooding pipeline, m
ω_{df}=cross-sectional area of one flooding hole, sq m
H_{df}=head, in mH_2O, in the unit at the flooding pipe.

Thus, the capacity of a fire pump must be

$$Q_f = Q_n n_n + Q_{ds} n_{ds} + Q_{fm} + Q_{df} \text{ cu m per hour} \qquad (17)$$

where Q_n and Q_{ds}=capacities required, respectively, for the operation of a fire nozzle and for a drencher and sprinkler head, cu m per hour
n_n and n_{ds}=number of fire nozzles, drencher and sprinkler heads
Q_{fm}=amount of water required to operate the foam-making installation, in cu m per hour
Q_{df}=water consumption of the deck flooding system, in cu m per hour.

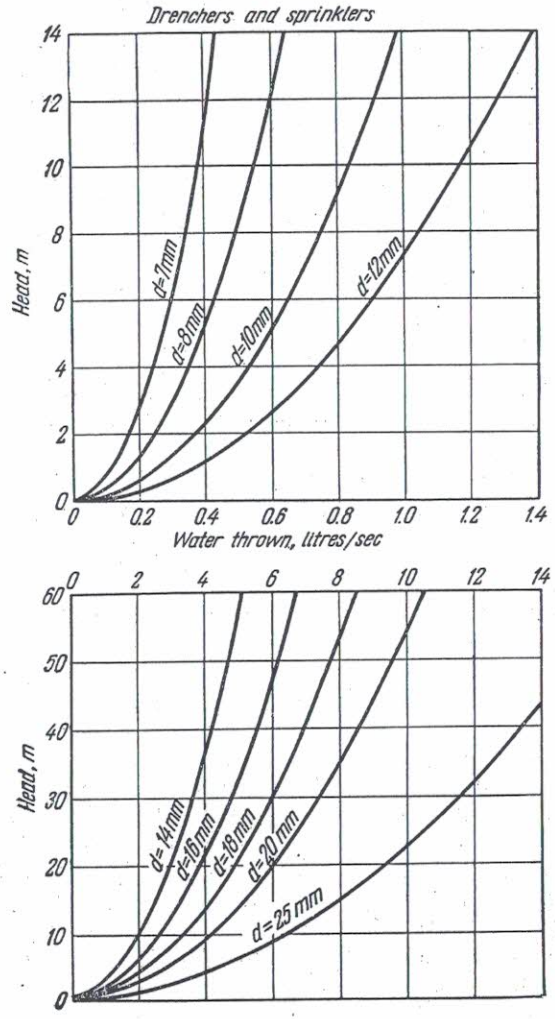


Fig. 4.

Fire pump capacities calculated by formula (17) should not be less than those given in Table 5.

Table 5

Type, size and purpose of ship	Minimum capacity and number of fire pumps
Sea-going passenger ships:	
Up to 500 gross tons	45 m³/hr, one
From 500 to 1,500 gross tons	45 m³/hr, two
From 1,500 to 4,000 gross tons	45 m³/hr, three, or two, 70 m³/hr each
From 4,000 gross tons and up	75 m³/hr, three
Self-propelled sea-going dry-goods carriers and oil tankers:	
Up to 1,000 gross tons	45 m³/hr, one
From 1,000 to 1,500 gross tons	45 m³/hr, two
From 1,500 to 4,000 gross tons	45 m³/hr, three, or two, 70 m³/hr each
From 4,000 gross tons and up	70 m³/hr, three, or two, 90 m³/hr each
Estuary passenger vessels:	
Length up to 70 m, incl.	30 m³/hr, one
Length over 70 m	30 m³/hr, two
Self-propelled estuary dry-goods carriers and oil tankers	30 m³/hr, one
Towing vessels, ice-breakers, rescue ships, fire-fighting boats and service ships, both sea-going and estuary	Subject to approval by the Shipping Register and Fire Department
Nonpropelled oil-transfer barges	30 m³/hr, one

Emergency Pumps

Emergency pumps are employed to pump water out of flooded compartments of the ship. The pertinent authorities do not stipulate data from which the capacities of such pumps could be calculated. In marine practice, however, these capacities usually range from 200 to 1200 cu m per hour or even higher.

If no special emergency pumps are available on the ship, the emergency system is connected through a bypass to the circulation pump of the main power plant.

(B) PUMP CLASSIFICATION ACCORDING TO PRINCIPLE OF OPERATION

The following types are distinguished:

1. Static head, or potential energy, pumps.
2. Dynamic head, or kinetic energy, pumps.

The first group of pumps, which mainly create a static head or potential energy, are of the positive displacement type and include reciprocating and rotary pumps.

In addition to the static head, the pumps of the second group impart considerable dynamic head, or kinetic energy, to the liquid being handled. The part of the kinetic energy (dynamic head) required to deliver the liquid to its destination is transformed into potential energy (static head) in special parts of the pump designed for this purpose. In other words, velocity head is converted into pressure head.

The second group includes centrifugal, propeller and jet pumps.

B. RECIPROCATING PUMPS

1-4. Classification, Typical Construction, Capacity and Nonuniform Flow

(A) CLASSIFICATION

Reciprocating pumps are ones in which the mechanical energy of the pump drive is converted into flow energy of the liquid being delivered, by means of a displacing element reciprocating in a cylinder.

The displacing element is called a piston or a plunger depending upon its construction. It imparts energy to the liquid by direct pressure exerted on the latter in the cylinder.

Reciprocating pumps can be classified according to:

Action—as single-acting and double-acting.

Pressure they develop—as low-pressure pumps (up to 5 kg per sq cm), medium-pressure (from 5 to 50 kg per sq cm) and high-pressure (over 50 kg per sq cm).

Capacity—as low-capacity (up to 20 cu m per h), medium-capacity (from 20 to 60 cu m per h) and high-capacity pumps (over 60 cu m per h).

Speed of the drive shaft, in rpm, or the number of piston or plunger double strokes per minute—as low-speed (up to 80), moderate-speed (from 80 to 150), high-speed (from 150 to 350) and extra-high-speed pumps (from 350 to 750).

Fluid being handled—as water, oil, petroleum, air pumps, etc.

Construction—as piston, plunger, simplex, duplex, triplex, vertical, inclined, horizontal, geared, gearless, etc.

Driving method—as power and direct-acting pumps. Power pumps are those in which the piston or plunger is driven through a crank mechanism. In direct-acting pumps the pump piston is driven by the piston rod of a steam engine on which both the engine and pump pistons are mounted.

Type of energy involved in their drive—as electric motor and steam engine pumps. In addition, some pumps are driven by internal-combustion engines.

(B) POWER RECIPROCATING PUMPS

1. Single-Acting Pump

Let us consider the function of the plunger pump of the power pumping installation shown in Fig. 5.

Here, plunger *13* is reciprocated in cylinder *12* by connecting rod *14* which is linked to the crankpin of flywheel *15* mounted on the motor or engine shaft. On the up stroke of plunger *13*, the volume not occupied by the liquid and plunger in cylinder *12* is increased by the amount of space freed by the ascending plunger. This increase in volume is accompanied by a drop in pressure in the cylinder, leading to the evolution of air dissolved in the liquid, as well as its vapour. When the pressure in cylinder *12* falls to a sufficiently low value, atmospheric pressure overcomes the weight of suction, or inlet, valve *21* and the pressure in the cylinder behind the valve, thereby lifting it from its seat. Thus, suction pipe *2* is connected through open gate valve *26* to cylinder *12*. Due to expansion, the air pressure in the suction pipe drops. As a result, the pressure p_a (atmospheric) forces the liquid through strainer *1* and flap valve *25* into suction pipe *2* where it begins to ascend. The strainer prevents mechanical impurities in the liquid from getting into the suction pipe. When the plunger finishes its up stroke (also called the suction stroke) and reaches its upper position, the maximum possible amount of air will have been drawn into the cylinder from the suction line. During a suction stroke the liquid is lifted to the height h in the suction pipe. In the down (discharge) stroke of the plunger, the pressure in cylinder *12* increases with the distance travelled by plunger *13*. When the pressure acting on suction valve *21* from the side of the cylinder becomes greater than that on the suction line side, the valve returns to its seat. The pressure in cylinder *12* and, consequently, under discharge valve *22* continues to increase upon further downward movement of the plunger. Valve *22* will be raised from its seat when this pressure becomes sufficient to overcome

the pressure acting on the top of the valve that is due to the pressure in discharge line *9*. This will connect cylinder *12* to the discharge line. During the rest of the down stroke, the plunger, occupying more and more space in the cylinder, forces air out of the cylinder and

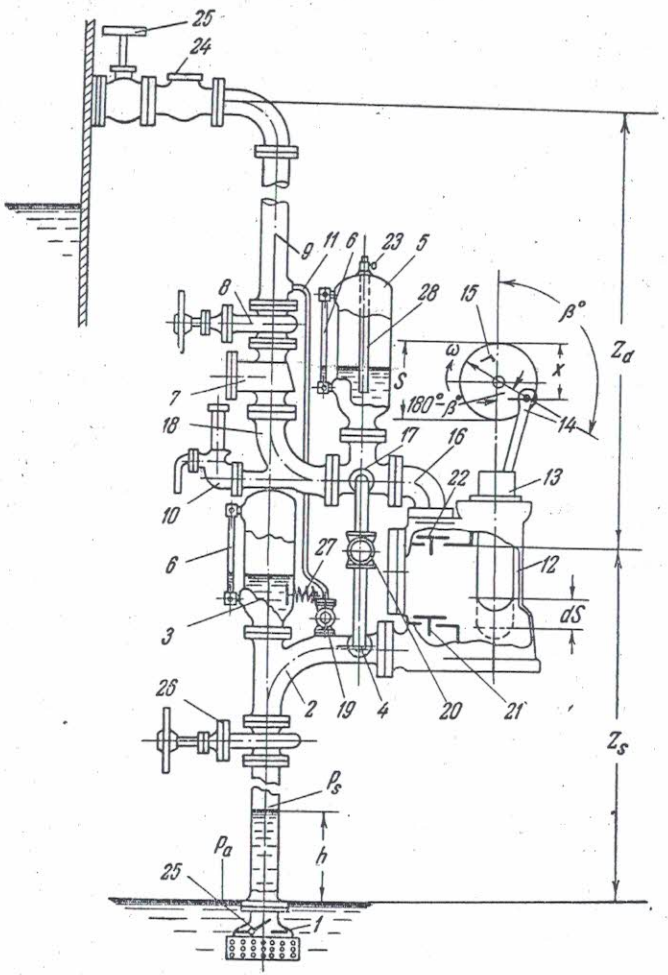


Fig. 5.

into the discharge line where it passes through elbow *16*, tee *17*, elbow *18*, valve *7*, gate valve *8*, swing-check valve *24*, screw-down nonreturn valve *25*, and overboard. Nonreturn valve *7*, gate valve *8* and swing-check valve *24* are installed in the line to prevent sea water

from entering the discharge line through the sea opening. The discharge stroke is followed by a new suction stroke and the cycle is repeated.

In the beginning, only air from the suction pipe enters the cylinder and the column of liquid rises in the pipe. Then a mixture of air and the liquid is drawn into the cylinder and, finally, only the liquid is being pumped. This means that the pump has begun normal operation.

Pumping installations are equipped with a vacuum chamber *3*, having a gauge glass *6* and air valve *27*, to equalize and increase the suction during the plunger stroke.

Since a single-acting pump discharges liquid only once in two strokes of the plunger (or piston) the liquid will be delivered intermittently to its destination. Discharge air chamber *5* is installed in the discharge line to obtain continuous and more uniform delivery. Chambers *3* and *5* have gauge glasses *6* and try cock *23* connected to tube *28*. High pressure in the discharge air chamber leads to increased dissolution of its air in the pumped liquid. The air is replenished by means of air valve *27* which admits air from the atmosphere into chamber *3* from where it passes through cylinder *12* to chamber *5*.

Relief valve *10* prevents the pressure in the discharge line from increasing beyond the normal value. The amount of liquid entering discharge line *9* is regulated by controlling the flow through bypass valve *20* which is mounted in pipe *4* leading from tee *17* to the suction pipe *2*.

Pipe *11* with valve *19* connects the discharge and suction lines and serves to fill the latter with liquid before starting the pump. This is called priming. Foot valve *25* prevents the liquid from running out of the suction pipe through strainer *1* during priming.

Let us find the height to which the liquid is lifted in the suction pipe during the first strokes of the plunger while the air is being removed from the suction pipe and cylinder.

The following equations are true for any position of the plunger after the cylinder is connected to the suction pipe:

$$p_a = p_s + \gamma h \quad \text{kg per sq m or } h = \frac{p_a - p_s}{\gamma} \text{ m} \qquad (18)$$

where p_s = pressure in the suction line and cylinder, kg per sq m, after they are connected through the suction valve
h = height to which the liquid is lifted in one suction stroke of the plunger, m.

On the other hand, according to Boyle's or Mariotte's law, we can write for the extreme upper position of the plunger that

$$V_p p_p = (V_p + FS)\, p_s$$

where V_p = volume, cu m, of the suction pipe not occupied by liquid before it is connected to the cylinder through the suction (inlet) valve
p_p = pressure, kg per sq m, in the suction pipe before it is connected to the cylinder through the suction valve
S = plunger stroke, m
F = cross-sectional area of the plunger, sq m.

It follows that

$$p_s = \frac{V_p p_p}{V_p + FS} \qquad (19)$$

Combining equations (18) and (19) we obtain

$$h = \frac{p_a - p_s}{\gamma} = \frac{p_a V_p + p_a FS - V_p p_p}{(FS + V_p)\gamma} = \frac{p_a}{\gamma} - \frac{V_p p_p}{(FS + V_p)\gamma} = \frac{1}{\gamma}\left(p_a - \frac{1}{\frac{FS}{V_p p_p} + \frac{1}{p_p}}\right) \text{ m} \qquad (20)$$

Since, during the first up stroke of the plunger, the suction pipe is filled with atmospheric air at the same pressure as the air outside the pipe, i.e.,

$$p_p = p_a$$

the height to which the liquid is lifted in the first suction stroke of the plunger will be:

$$h = \frac{p_a - p_s}{\gamma} = \frac{V_p p_a + p_a FS - V_p p_a}{(FS + V_p)\gamma} = \frac{p_a}{\left(1 + \frac{V_p}{FS}\right)\gamma} \text{ m} \qquad (21)$$

It is evident from equation (20) that the greater the volume freed by the plunger in one stroke and the lesser the density γ of the liquid being pumped, the higher the liquid is lifted in the suction pipe in one suction stroke of the plunger.

Since the density of water $\gamma \approx 1{,}000$ kg per cu m, it follows from equation (21) that

$$h = \frac{10{,}000}{\left(1 + \frac{V_p}{FS}\right) 1{,}000} = \frac{10}{1 + \frac{V_p}{FS}} \text{ m} \qquad (22)$$

In one suction and one discharge stroke the liquid is raised to the geometric height z_s, or suction lift, and the height z_d, or discharge head. Due to the variable speed of the plunger (or piston) and the alternation of suction and discharge strokes, reciprocating pumps cannot produce a steady flow of liquid in the piping, and the discharge varies within each revolution of the pump shaft (two plunger strokes).

We can now derive the law according to which the discharge varies and find the displacement value for a single-acting power pump.

The pump discharge dQ for the infinitesimal movement dS of the plunger (Fig. 6) that takes place at the velocity c during the infinitesimal period of time dt is

$$dQ = FdS = Fcdt \tag{23}$$

The distance x travelled by the plunger from its upper position, disregarding the effect of the angularity of the connecting rod, dur-

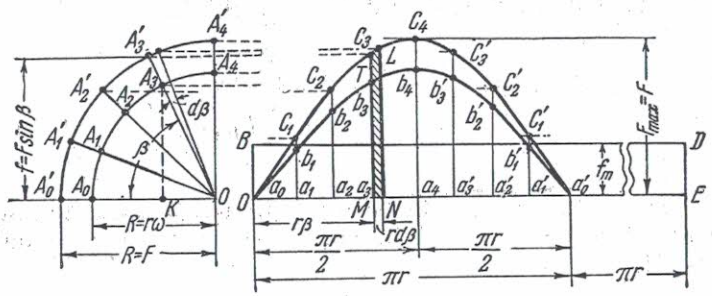


Fig. 6.

ing crank rotation through angle β° can be calculated from the formula

$$x = r + r\cos(180° - \beta°) = r - r\cos\beta° = r(1 - \cos\beta°) \tag{24}$$

The instantaneous velocity c of the plunger after the crank has rotated through the angle β° will be

$$c = \frac{dx}{dt} = \frac{d}{dt}[r(1 - \cos\beta°)] = r\sin\beta°\frac{d\beta°}{dt} = r\omega\sin\beta° \tag{25}$$

Thus, the plunger velocity at a given crank position is represented by the ordinate of a harmonic curve and depends upon the angle through which the crank has turned from its upper dead point, taken as the datum.

Let us plot the harmonic curve of the variation in plunger velocity during one stroke, i.e., when the angle β° of crank rotation varies from 0 to 180° (Fig. 6). First we draw a quarter of a circle with a radius $R = r\omega$ in any suitable scale. Extending the base line A_0O we draw the straight line $OE = 2\pi r$ in a suitable scale and then lay off the length a_0a_0' equal to the length of the semicircle πr, described by the crankpin in its rotation through the angle β°=180°.

Next we divide the quarter of a circle into $n=4$ equal parts, denoting the points as A_1, A_2, A_3 and A_4, and the line a_0a_0' into $2n=8$ equal parts, denoting the points as a_1, a_2, a_3, a_4, a_3', a_2' and a_1'. Then we erect perpendiculars from the latter set of points. Projecting horizontal lines from points A_1, A_2, A_3 and A_4 to their intersections with the perpendiculars, we obtain points b_1, b_2, b_3, b_4, b_3', b_2' and b_1' lying on the harmonic curve, which we can now draw. The ordinates of the curve demonstrate the variation in the velocity of the plunger during a single stroke. As a matter of fact, any ordinate of the curve can be obtained by means of an accessory circle. Thus

$$A_3K = b_3a_3 = r\omega\sin\beta°$$

The plunger attains its maximum velocity $r\omega$ at the angle β°=90°.

Pump discharge, or delivery, during crankpin rotation from any position determined by the angle β° through the infinitesimal angle $d\beta°$, according to equations (23) and (25), will be

$$dQ = Fr\omega\sin\beta°\,dt = Fr\sin\beta°\,d\beta° \tag{26}$$

After dividing the angle of rotation of the crankpin, corresponding to the discharge stroke, into infinitesimal angles $d\beta°$ and summing up the discharges corresponding to these angles, we obtain the delivery Q_β for any angle β° of crankpin rotation:

$$Q_\beta = \int_0^{\beta°} Fr\sin\beta°d\beta° = Fr\,[-\cos\beta°]_0^{\beta°} = Fr(1 - \cos\beta°) \tag{27}$$

The integral of equation (27) represents the area bounded by a section of the harmonic curve and its base line. The harmonic curve of pump discharge can be plotted as described above if the cross-sectional area F of the plunger (or piston) is taken as the radius R of the accessory circle. Here, each ordinate of the curve, corresponding to a definite angle of crankpin rotation β° is equal to $F\sin\beta°$, and the distance travelled by the crankpin in its rotation through the angle $d\beta°$ is represented by the line MN, equal to $rd\beta°$. Therefore, the hatched area $MNTL$, equal to $Fr\sin\beta°d\beta°$, will be the pump discharge for rotation of the crankpin through the angle $d\beta°$.

The discharge when the crankpin rotates from the angle β°=0° to β°=180°, i.e., during one stroke of the plunger will be

$$Q = \int_{\beta°=0°}^{\beta°=180°} Fr\sin\beta°d\beta° = Fr\,[-\cos\beta°]_{\beta°=0°}^{\beta°=180°} =$$
$$= Fr + Fr = 2Fr = FS \tag{28}$$

This value is generally called the displacement of the pump.

The theoretical hourly capacity of a single-acting pump with a shaft speed of n rpm (or n double strokes of the plunger per minute) is

$$Q_t = 60\,FSn \text{ cu m per hour} \tag{29}$$

while the actual capacity is

$$Q = 60\eta_v FSn \text{ cu m per hour} \tag{30}$$

It is evident that the variation in plunger velocity and in discharge of the liquid into the delivery line can be represented by the same harmonic curve. The absolute velocity and discharge values can be found, however, only on the basis of a different scale in each case.

The next item of interest is the variation in the flow of a single-acting pump.

Since a single-acting pump effects discharge during one stroke of the plunger and draws the liquid into the cylinder on the second stroke, its flow curve will have the form shown in Fig. 6.

If a single-acting pump discharged liquid in a uniform flow and during both strokes of the plunger its displacement could be represented by the area of the rectangle *OBDEO* which is equal to the area bounded by the loop of the harmonic curve and the base line. In this case the ordinates of the line *BD*, equal to f_m, would represent in one scale, the constant velocity of the pump plunger, and in another scale, the pump discharge per second.

Let us call the ratio of the maximum discharge per second, q_{max}, to the mean discharge per second, q_m, during two strokes of the plunger the variability of liquid flow δ_v. This is the same as the ratio of the maximum plunger velocity, c_{max}, to the constant plunger velocity, c_c. Thus

$$\delta_v = \frac{q_{max}}{q_m} = \frac{c_{max}}{c_c} = \frac{F_{max}}{f_m} \tag{31}$$

On the basis of equation (28) and Fig. 6 we can write

$$SF = 2rF = 2\pi r c_c = 2\pi r f_m$$

whence

$$f_m = \frac{2rF}{2\pi r} = \frac{F}{\pi}$$

Since in a single-acting pump

$$c_{max} = F_{max} = F$$

then

$$\delta_v = \frac{F}{\frac{F}{\pi}} = \pi = 3.14 \tag{32}$$

2. Differential Plunger Pump

If it becomes necessary to achieve a more uniform flow of the liquid, i.e., to deliver the liquid during both strokes of the plunger without increasing the pump capacity, a single-acting pump with a differential plunger is employed. The principle of this pump is illustrated in Fig. 7. Upon movement of plunger *1* to the right, liquid enters cylinder chamber *5* through suction pipe *2* and inlet valve *4*. In the plunger stroke to the left this liquid is forced through discharge valve *9* from where part of the liquid passes through bypass

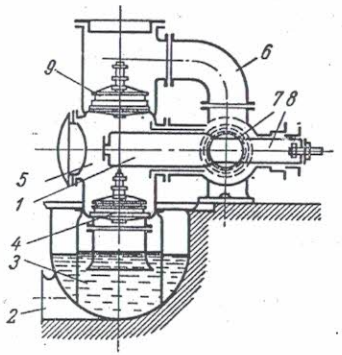


Fig. 7.

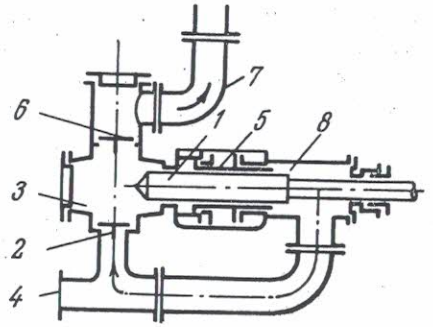


Fig. 8.

6 and fills cylinder chamber *7*, while the rest, of a volume equal to that of the effective part of plunger rod *8*, is delivered into the discharge line through the discharge connection. In the next stroke of the plunger to the right, the volume of liquid that entered chamber *7* will be forced into the discharge line and suction will take place in chamber *5*. Thus, the pump delivers liquid during both strokes of the plunger.

If we denote the cross-sectional area of the plunger rod as f_r the displacement of a differential pump on the plunger stroke to the right, using the same notation as before, will equal $(F-f_r)\,S$ while on the stroke to the left it will equal $f_r S$. Obviously, the pump displacement for both strokes of the plunger will be the sum of the displacements for each stroke, or

$$(F-f_r)S + f_r S = FS$$

In other words, the capacity of a single-acting pump with a differential plunger or piston is equal to that of a single-acting pump with an ordinary plunger or piston. The flow in the discharge line, however, will be twice as uniform in the differential type.

A suitable arrangement of the valves in the cylinder of a differential pump enables the liquid to be drawn into the cylinder during both strokes and discharged on one stroke. Such pumps find application when a more uniform flow is required in the suction line than can be achieved with the ordinary single-acting pump. They are usually installed in long and circuitous suction lines. This type of arrangement is shown schematically in Fig. 8.

Upon travel of plunger *1* to the left, liquid is drawn into chamber *8* of the cylinder and discharged from chamber *3* through valve *6*. As the plunger travels to the right only suction takes place in chamber *3*. A part of the liquid is drawn in from chamber *8* through inlet valve *2* and the remainder from the suction line *4*. The plunger is inside-packed with stuffing box *5*.

If the plunger rod is designed so that its cross-sectional area is one half of that of the plunger then the discharge of the differential plunger pump shown in Fig. 7 will be the same for both strokes. This same feature enables a pump of the type shown in Fig. 8 to draw the same amount of liquid into the cylinder in each stroke.

The flow curve of a differential pump is similar to that of a double-acting pump, as will be seen below.

3. Double-Acting Pump

A double-acting pump is one in which the amount of liquid discharged during two strokes of the plunger or piston is approximately equal to two effective volumes of the cylinder. In calculating pump displacement it is necessary to use a coefficient whose value depends upon the type of pump, stated in terms of the number of discharging strokes per pump shaft revolution. The discharge stroke coefficient, as we shall call it, is equal to unity for a single-acting simplex (single-cylinder) pump. The coefficient is equal to 2 for the double-acting simplex pump illustrated schematically in Fig. 9.

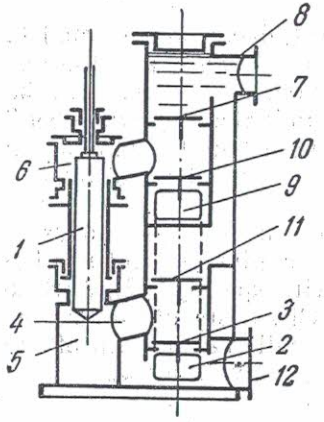


Fig. 9.

On the down stroke of plunger *1* liquid is drawn into the upper valve chest through inlet connection *12* and ports *2* and *9*. From here it passes into upper chamber *6* of the cylinder through suction valve *10*. At the same time, liquid from lower chamber *5* of the cylinder, drawn into the chamber on the preceding stroke, is forced through connection *4* and discharge valve *11* into the discharge chamber and further through connection *8* to the discharge line. The plunger reverses after it reaches its lower dead point and begins to force out the liquid, drawn into the upper chamber of the cylinder on the preceding stroke, through discharge valve *7*, and simultaneously draws liquid into the lower chamber through valve *3*.

The theoretical capacity of a double-acting pump is equal to

$$Q_t = (F - f_r)S + FS = 2FS - f_r S$$

assuming that

$$\frac{F + (F - f_r)}{2} = F - \frac{f_r}{2} = F_m$$

then

$$Q = 2 \times 60 \eta_v F_m S n = 120 \eta_v F_m S n \text{ cu m per hour} \qquad (33)$$

In tentative calculations the value $f_r S$ can be neglected.

The reduction in volume of the cylinder chamber, through which the rod passes, must be taken into account in plotting capacity

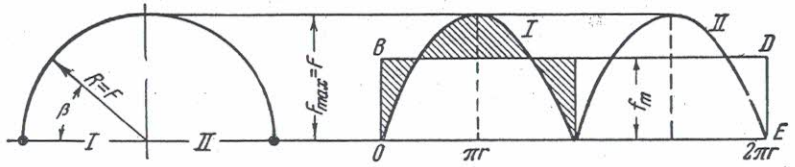


Fig. 10.

curves for double-acting pumps. This is done by reducing the piston (or plunger) area, F, by the cross-sectional area of the rod, f_r. Based on the lag or "phase difference" of one stroke, i. e., 180° of crankpin rotation, between discharge by one side of the plunger and by the other side, the capacity curve can be represented as in Fig. 10.

The discharge of a double-acting pump per revolution of the drive shaft (neglecting the volume of the rod) is equal to twice the displacement, i.e., the volume swept through, or displaced, by the piston or plunger in a single stroke. If a mean capacity curve is to be plotted, its ordinate will be proportional to the mean discharge:

$$f_m = \frac{2FS}{2\pi r} = \frac{4Fr}{2\pi r} = \frac{2F}{\pi}$$

Then the variability of liquid flow will be

$$\delta_v = \frac{c_{max}}{c_m} = \frac{F_{max}}{f_m} = \frac{F}{\frac{2F}{\pi}} = 1.57 \qquad (34)$$

It follows that the variability of flow in double-acting pumps is only one half that of single-acting pumps.

The same effect may be achieved by combining two identical single-acting pumps driven from a single crankshaft with the crankpins arranged at 180° to each other. The two pumps have a common inlet and a common discharge connection. The discharge stroke of one pump begins at the end of the discharge stroke of the other. Such an arrangement is called a duplex single-acting pump.

4. Triplex Single-Acting Pump

A triplex single-acting pump is employed to obtain increased capacity and more uniform flow. It is, in effect, three single-acting simplex pumps driven by the crankpins *1*, *2* and *3* (Fig. 11) of a

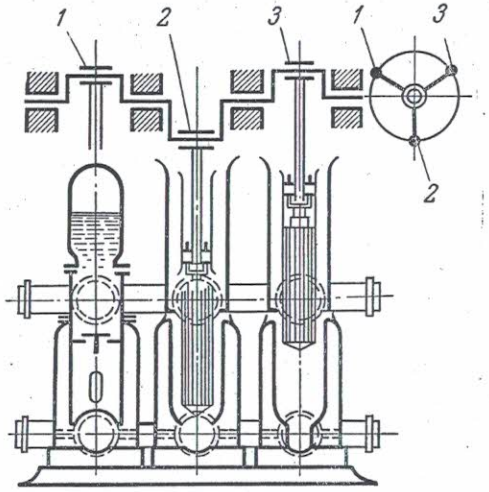


Fig. 11

crankshaft. These crankpins should be arranged at angles of 120° to each other and the pumps have a common suction line and a common discharge line. This pump has a discharge stroke coefficient of three.

The capacity of a triplex pump is three times greater than that of a simplex single-acting pump:

$$Q = 3 \times 60\eta_v FSn = 180\eta_v FSn \text{ cu m per h} \qquad (35)$$

The capacity curve of the triplex pump can be plotted by shifting the flow curve of the second single-acting cylinder by 120° and the

third by 240°, in reference to the first cylinder, from the datum of crankpin rotation (Fig. 12).

The ordinates at the points of intersection of the first cylinder flow curve with the second, the second with the third and the third with the first can be found easily from the angles of crankpin rotation, corresponding to these points, which are 30°, 150° and 270°. Thus

$$f = R \sin \beta^\circ = F \sin \beta^\circ = F \sin 30^\circ = 0.5F$$

The overlapping discharges of two cylinders at each moment during the cycle do not allow the total capacity of the pump to drop

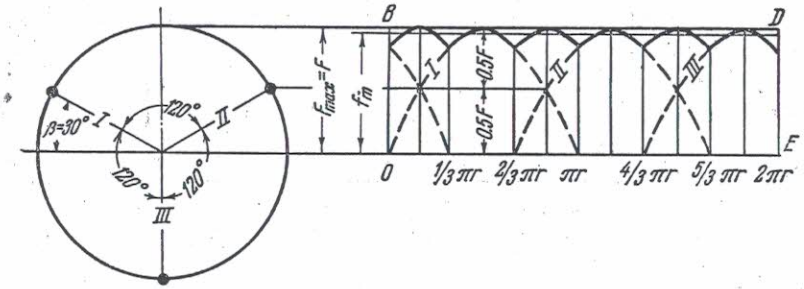


Fig. 12.

to zero as observed in the simplex single- and double-acting and in the duplex single-acting pumps. The maximum ordinate of the curve $F_{max}=0.5F+0.5F=F$.

Since the pump displacement per shaft revolution is equal to $3FS$ and the maximum ordinate of the curve is $F_{max}=F$

$$f_m = \frac{3FS}{2\pi r} = \frac{6Fr}{2\pi r} = \frac{3F}{\pi}$$

and

$$\delta_v = \frac{c_{max}}{c_m} = \frac{F_{max}}{f_m} = \frac{F}{\frac{3F}{\pi}} = 1.047 \qquad (36)$$

5. Duplex Double-Acting Pump

This pump consists of two double-acting cylinders in which the piston of one cylinder lags behind or leads the other by one half of a stroke. Such a pump has a discharge stroke coefficient of four.

The crankshaft which drives the pistons or plungers of this pump has two crankpins arranged at 90° to each other. The capacity of

such a pump is twice that of a simplex double-acting pump. Thus

$$Q = 2 \times 120\eta_v FSn = 240\eta_v FSn \text{ cu m per h} \quad (37)$$

The capacity curve of a duplex double-acting pump is of the form shown in Fig. 13. The ordinates at the intersections of the harmonic curves for the separate cylinder ends are

$$f = R \sin\beta^\circ = F \sin\beta^\circ = F \sin 45^\circ \approx 0.7F$$

Therefore, F_{max} is equal to 1.4 F. The displacement of the pump per

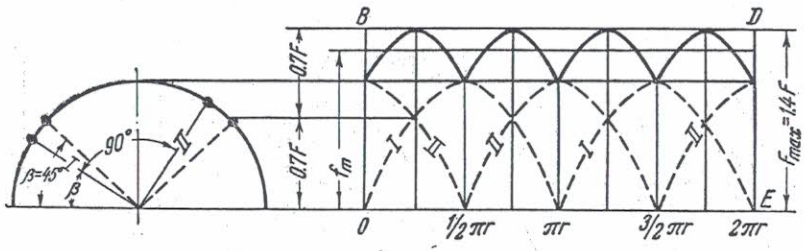


Fig. 13.

shaft revolution being $4FS$ (neglecting the volume of the rod), we can write

$$f_m = \frac{4FS}{2\pi r} = \frac{8Fr}{2\pi r} = \frac{4F}{\pi}$$

$$\delta_v = \frac{c_{max}}{c_m} = \frac{F_{max}}{f_m} = \frac{1.4F}{\frac{4F}{\pi}} = 1.11 \quad (38)$$

It follows that a duplex double-acting pump has less uniform flow than a triplex pump. Therefore, triplex single-acting pumps are applied in installations requiring highly uniform flow and a moderate capacity (for example, boiler feed pumps).

6. *Multiple-Cylinder Pumps*

Multiple-cylinder pumps with higher discharge stroke coefficients are obtained by combining single- or double-acting pumps into a single unit, and are employed to increase capacities and to save space.

The capacity of a pump with a discharge stroke coefficient of k is

$$Q = 60k\eta_v FSn = 60k\eta_v \frac{\pi D^2}{4} Sn = 47.1\eta_v kD^2 Sn \text{ cu m per h} \quad (39)$$

where D = cylinder bore, m.

If the mean velocity of the piston $c_m = \frac{2Sn}{60}$ m per second, then $Sn = 30c_m$ and

$$Q = 47.1\eta_v kD^2 \times 30c_m = 1413\eta_v kD^2 c_m \text{ cu m per h} \quad (40)$$

Pump capacity can also be expressed in terms of the value ψ which is the ratio of the piston stroke to the cylinder bore. Since $\psi = \frac{S}{D}$ and $S = \psi D$, it follows from equation (39) that

$$Q = 47.1\eta_v k\psi D^3 n \text{ cu m per h} \quad (41)$$

The speed in double strokes of the piston per minute and the piston stroke-to-cylinder bore ratios of the most extensively used pumps are given in Table 6.

Table 6

Type	Number of double strokes per min	$\psi = \frac{S}{D}$
Low-speed power pumps	from 40 to 80	2.5 to 2
Moderate-speed power pumps	„ 80 „ 150	2 „ 1.2
High-speed power pumps	„ 150 „ 350	1.2 „ 0.5
Extra-high-speed power pumps	„ 350 „ 750	0.5 „ 0.2
Direct-acting pumps	„ 25 „ 130	1.75 „ 1.0
Hand pumps	„ 20 „ 45	2 „ 0.8
Hand fire pumps	„ 30 „ 60	3.5 „ 2.5

The required discharge stroke coefficient, characterizing the number of cylinders and the working cycle of a pump selected for installation, depends upon the desired uniformity of flow and the purpose of the pump.

The following data demonstrate the variability of liquid flow

Discharge stroke coefficient k	δ_v
Five	1.016
Six	1.047
Seven	1.008
Eight	1.026
Nine	1.005

(C) DIRECT-ACTING PUMPS

Direct-acting pumps have no crank mechanism and the piston rod of the steam cylinder is rigidly linked by a coupling to the piston rod of the hydraulic cylinder.

Direct-acting pumps are usually of the simplex or duplex double-acting type. A duplex double-acting pump is shown schematically in Fig. 14.

Pistons *1* and *2* of steam cylinders *3* and *4* drive pistons *5* and *6* of hydraulic cylinders *7* and *8*.

Each end of the cylinders has an inlet valve (*9*, *10*, *11* and *12*) and a discharge valve (*13*, *14*, *15* and *16*). The liquid being pumped flows to inlet valve chest *17* through connection *18* and is delivered to the discharge line from discharge valve chests *19* and *20* through connection *21*. Valve chests *19* and *20* are connected by channel *22* and ports *23* and *24*.

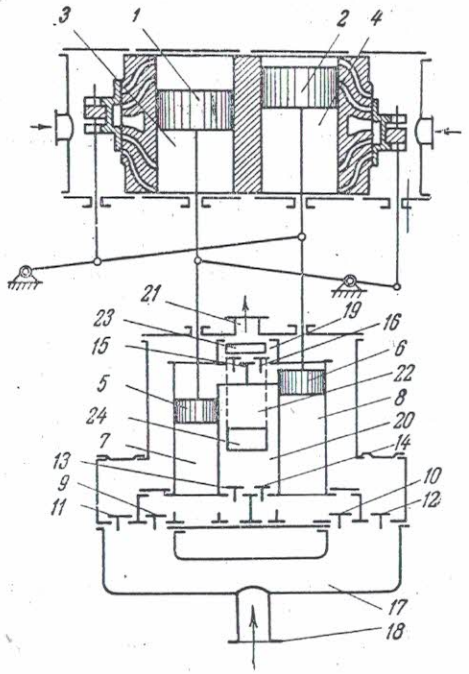


Fig. 14.

The capacity of direct-acting pumps is calculated by the same formulas that were used for power pumps.

The variation in the piston velocity of direct-acting pumps and, consequently, in displacement during two strokes of the piston does not follow the law derived previously for power pumps. The piston velocity of a direct-acting pump varies with the pressure in the steam cylinder. The piston is accelerated in the initial period when the available force exceeds the resistance to movement which increases with the piston velocity. At a certain point, almost immediately after the beginning of the stroke, the increased resistance balances the excess force and steady motion is established.

This is clearly evident from the piston travel vs time diagram (x vs. t) shown in Fig. 15. The distance that the piston travels in the infinitesimal period of time dt will obviously be dx. The piston velocity will be

$$c = \frac{dx}{dt}$$

Denoting by $\varphi°$ the angle between the tangent to the curve ab, representing the piston velocity c, and the line representing dt, we can write

$$c = \frac{dx}{dt} = \tan \varphi° \qquad (42)$$

This means that the tangent of the angle made by the tangent to the curve ab at any point and the axis of abscissas is the piston velocity at the given moment.

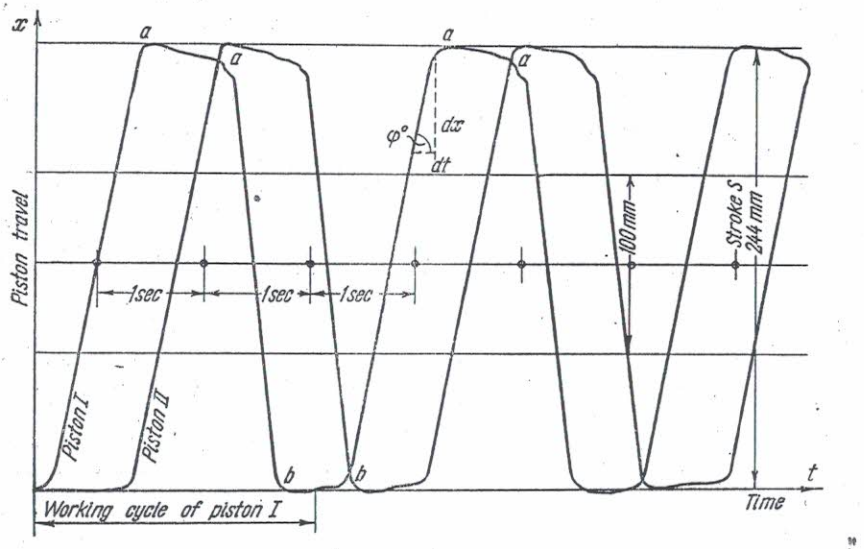


Fig. 15.

The diagram (Fig. 15) shows that $\varphi°$ is constant over almost the whole stroke except for the small sections a-a at the beginning and b-b at the end. It also shows that the initial acceleration and final deceleration are of short duration. This leads to the conclusion that the piston speed and, consequently, pump discharge remain constant during almost the whole stroke. Thus, a direct-acting pump provides the most uniform flow of all the reciprocating pumps considered so far.

The sections a-a and b-b refer to intervals in which the piston is slowing down, standing still and beginning a new stroke. When one piston begins to decelerate at the end of its stroke in a duplex pump the other piston reaches its maximum velocity, thereby achieving almost uniform flow of the liquid in the discharge line.

A very simple instrument is used to draw a diagram of the type shown in Fig. 15. It consists of a drum with paper attached to it, mounted with its axis parallel to the pump axis. The pump is driven at a constant speed and a pencil, attached to the piston rod, draws the curve.

A piston velocity variation curve and, therefore, a pump discharge curve can be constructed by graphical integration.

If time is plotted along the axis of abscissas in Fig. 15 and pump discharge along the axis of ordinates then the area under the curve will be equal to the pump displacement in some scale. The theoretical variability of flow of direct-acting pumps equals unity while the actual value ranges from 1.05 to 1.10.

1-5. Construction of Reciprocating Power Pumps

Shipboard power pumps may be either of the:

(1) attached type, which are driven by the main or auxiliary engines of the ship's power plant, or

(2) self-contained type, which have an individual motor or engine drive.

Fig. 16 illustrates the construction of an attached triplex pump whose plungers *1* are driven by means of connecting rods *2* and eccentric strap *3* from eccentric sheave *4* mounted on the end journal *5* of the main steam engine.

Each cylinder has a separate suction connection *6*, inlet valve *7*, discharge valve *8* and discharge air chamber *9* with test cock *10*. Water is drained from the cylinders by removing plugs *11*. Stuffing box *12*, kept watertight by the pressure applied to the packing by means of gland *13* and nuts *14*, prevents leakage from the cylinder on the discharge stroke and loss of vacuum in the suction stroke. The cylinders have the following functions: the left-hand cylinder serves the drainage system, the middle cylinder serves the boiler feed system and the right-hand cylinder—the sanitary system.

A triplex differential fire, drainage and ballast pump is shown in Fig. 17. The head of the combined piston and plunger is of the valved type. Such valved pistons are also called buckets.

The suction box serves as the foundation plate of this pump. It contains the seats *2* of the disk-type inlet valves *24* held down by springs *11*. The latter bear against retainer *23* which is secured by nut *12*. The valved differential pistons *4* reciprocate in liners *3* fitted into cylinder block *22*. The pistons are reciprocated by connecting rods *6* which are linked to crankshaft *16*. The crankshaft has three crankpins at 120° and on it is mounted worm wheel *17* driven by worm *18* which is linked to the shaft of the drive motor. The

valved pistons *4* have annular disk valves *13* loaded by springs that bear against retainers *7*.

Pins *5* and *27* link the piston to plunger *8* and connecting rod *6*. The plunger packing is contained in housing *9* and compressed by gland *10*. The crankshaft runs in bearings *26* which are mounted in uprights *14* and columns *15*. The bearings are lubricated by wick-feed oilers *25*.

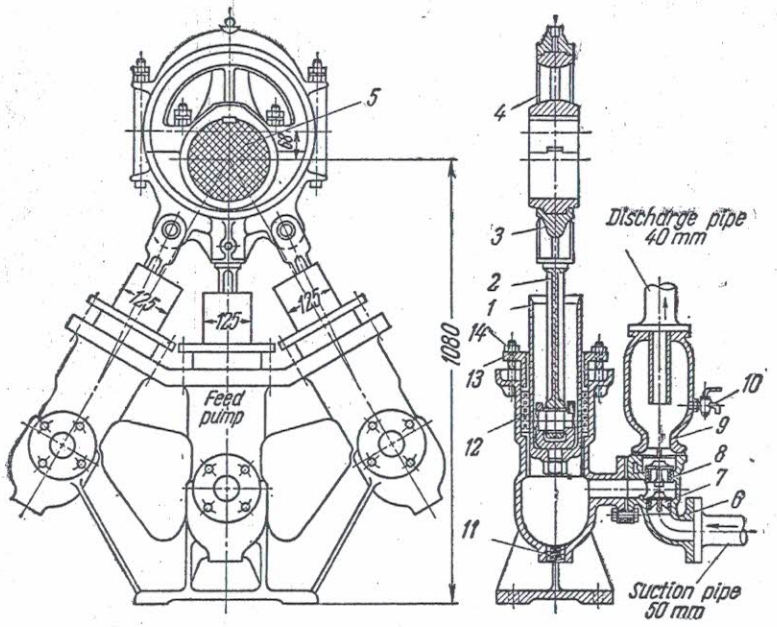


Fig. 16.

The oil in worm gear housing *19* is cooled by water from the discharge chamber passing through coil *20* and tube *29* to the suction chamber. The oil level in housing *19* is checked by means of gauge glass *37*.

Pressure gauge *30* is connected through tube *31* to discharge connection *32*, while a vacuum gauge is connected through tube *33* to the suction connection *35*.

The pump is equipped with two suction air chambers (called vacuum chambers) *28* and *36* and relief valve *21* which admits water from the discharge chamber to the vacuum chamber when the discharge pressure exceeds the permissible value. The vacuum chamber has an air valve *34*.

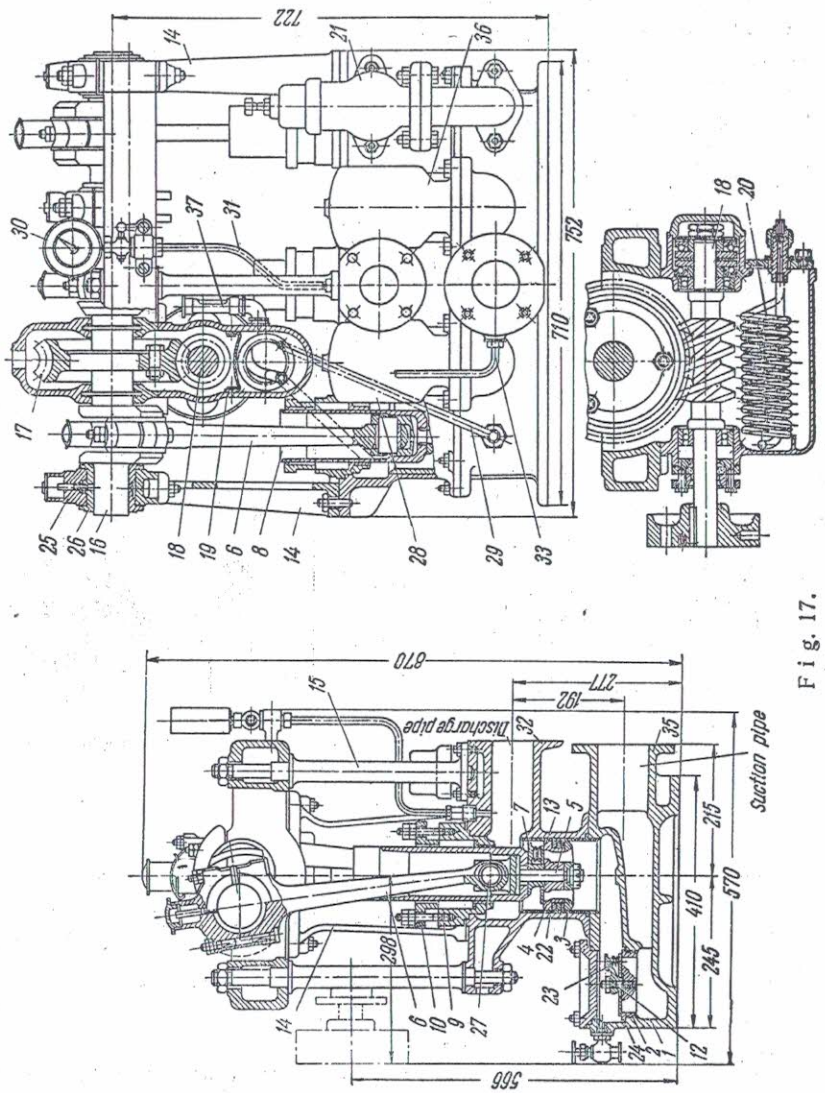


Fig. 17.

On the up stroke of the piston the liquid is drawn through connection *35* and inlet valve *24* into the lower end of cylinder *22*. On the down stroke the liquid in the lower end of the cylinder passes through the channels in the piston and valve *7* into the upper end of the cylinder. A part of this liquid remains in the cylinder and the remainder is forced into the discharge line by plunger *8*. On the next up stroke the rest of the liquid in the upper end of the cylinder is discharged. Thus the flow of this pump is as uniform as that of a triplex double-acting pump (in which the discharge stroke coefficient equals six) while the capacity is that of a triplex single-acting pump.

The capacity of reciprocating pumps is varied by changing the number of piston double strokes per minute or by bypassing liquid from the discharge to the suction line. The first is the more economical method and is accomplished in steam pumps by closing or opening the steam admission valve in the live steam line. The application of this method to pumps driven by electric motors involves the installation of devices for varying the motor speed. This is more expensive and complicates the installation and operation of the pump. Therefore, motor-driven pumps operate with a constant speed and their capacity is varied by bypassing the liquid as in the second method even though this is a less economical procedure.

A more economical method of varying power pump capacity is by changing the stroke of the pistons or plungers. This feature has been incorporated in the design of the single-acting triplex pump illustrated in Figs. 18*a* and 18*b*.

The pump is mounted in a closed housing together with drive motor *21*. Through a herringbone reducing gear, the motor drives crankshaft *1* which reciprocates rod *9* and plunger *8* by means of connecting rod *2* linked to rocker arm *3*. Slide block *4*, at the lower end of rocker arm *3*, slides in the curvilinear slot formed by guides *6* and shoe *5*. The latter can be swivelled in its trunnion bearings by the action of the piston in cylinder *7* which is controlled by the boiler water level. Each position of the piston in cylinder *7* corresponds to a definite inclination of shoe *5* and, therefore, to a definite stroke of plunger *8* and a corresponding displacement. The guiding slot in the shoe has a constant curvature along its entire length. Consequently, in the shoe position shown by full lines in Fig. 18*b*, the plunger stroke and the displacement will equal zero. The stroke will be maximum in the position shown by dotted lines. The three cylinders of the pump are arranged in block *10* and each of their plungers is reciprocated by a mechanism of the type described.

Each cylinder has an inlet valve *11* and discharge valve *12*, the chest of the latter being an extension of the cylinder. Liquid is drawn into the pump through hole *13* and discharged through hole *14*.

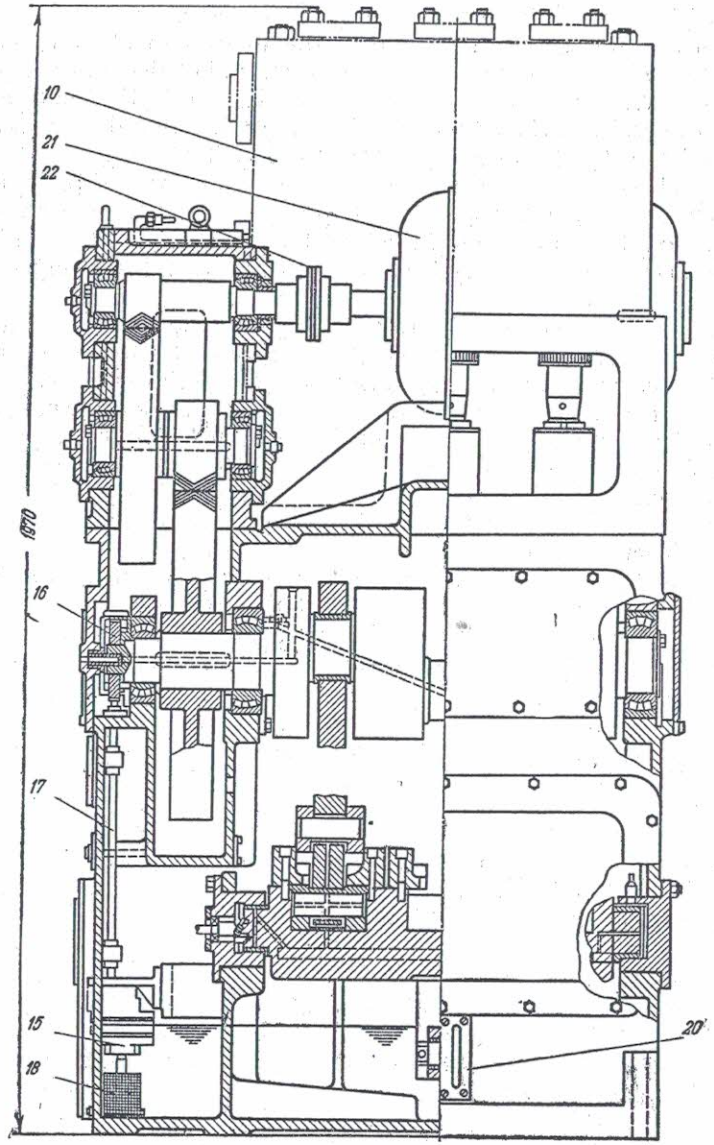


Fig. 18*a*.

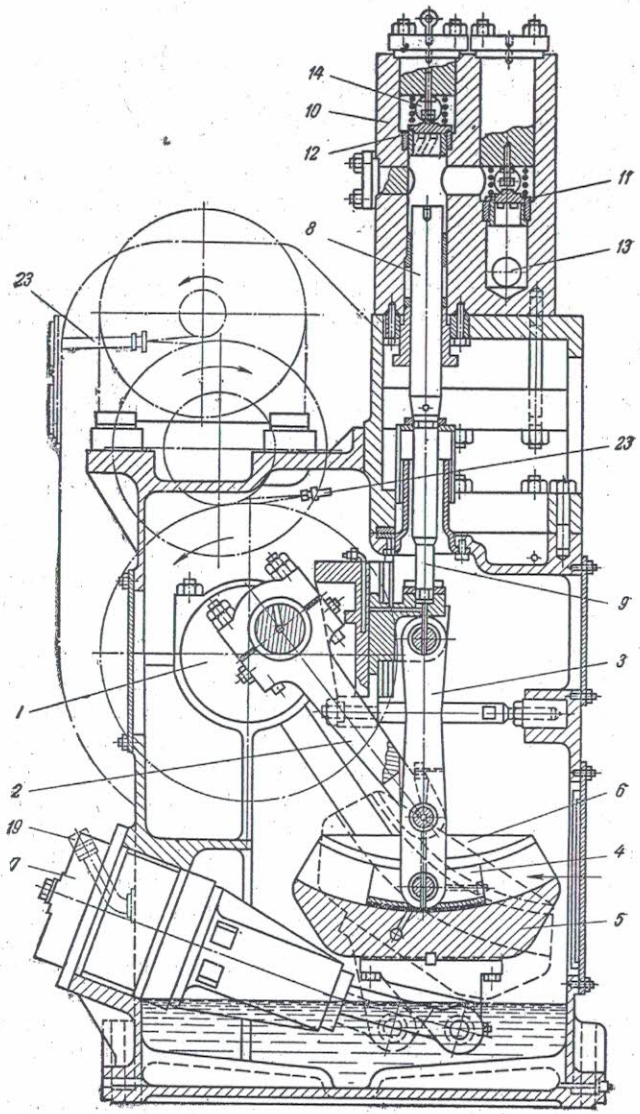


Fig. 18*b*.

The crankshaft runs in double-row self-aligning roller bearings as do the reducing gear shafts.

Oil is delivered to the friction surfaces of the pump parts under pressure by gear pump *15* which is driven from the crankshaft by gear *16*, a worm and shaft *17*. The gears are lubricated through connections *23*.

Lubricating oil is poured into the crankcase through pipe *19* and enters pump *15* through strainer *18*. The oil level is checked by means of sight gauge *20*. Flexible coupling *22* links motor *21* to the reducing gear.

The mechanical efficiency, η_m, of this pump varies with the plunger stroke (and pump capacity). If the maximum stroke is denoted as *S*, then

at a stroke *S*	$\eta_m \approx 0.80$
" " " 0.75 *S*	$\eta_m \approx 0.75$
" " " 0.5 *S*	$\eta_m \approx 0.67$
" " " 0.25 *S*	$\eta_m \approx 0.50$

1-6. Construction of Direct-Acting Steam-Driven Reciprocating Pumps

Due to their simple construction, dependability, small floor space occupied, light weight, highly uniform flow and high volumetric efficiency, direct-acting steam pumps have found wide application on ships. A disadvantage of these pumps is the high steam consumption which averages from 40 to 50 kg per ihp (indicated horse power) and exceeds that of power pumps. However, if the exhaust steam is utilized for miscellaneous shipboard needs, including the preheating of the boiler feed water, their many advantages make their use on steam ships highly expedient. In principle, the water end of direct-acting pumps in no way differs from those previously described.

Direct-acting pumps are available in vertical and horizontal designs and may be either simplex or duplex.

According to the method used to actuate the valve gear at the steam end direct-acting pumps may be classified into three types:

(1) pumps with positive-action mechanically operated valve gear,
(2) pumps with positive-action semiautomatic valve gear, and
(3) pumps with automatic valve gear.

(A) PUMPS WITH POSITIVE-ACTION MECHANICALLY OPERATED VALVE GEAR

1. Valve Gear Construction and Principles

Pumps with positive-action mechanically operated valve gear are of the duplex type having two steam and two hydraulic cylinders, all with double-acting cycles of operation, so that the discharge stroke coefficient equals four (Fig. 19). The distinguishing feature of these pumps is that the steam valve of one cylinder is operated by the piston rod of the second adjacent cylinder. The valve gear mechanism of such pumps, made by different manufacturers, may differ in details but the principle is the same as that devised by Henry R. Worthington in 1884.

Pistons *1* of steam cylinders *2* are advanced by steam admitted into the cylinders from steam chest *3*.

Rods *4* of the steam cylinders are linked to piston rods *5* of the water cylinders *6* by means of crossheads *10*.

Double-piston valves *7* of steam cylinders *2* are connected by valve stems *8* and connecting links *9* to a system of rocker arms pivoted in bearings. The steam cylinders have blow-down cocks *12* and cushion valves *11* for regulating the degree of cushioning (see page 64 and Fig. 22). In addition to piston valves, common slide valves without lap are often used in duplex pumps. The operation of the valve gear of such pumps is illustrated in Fig. 20.

The seat in the chest of each valve has five ports. The inner passages *2*, *6*, *8* and *11* release the exhaust steam from the cylinders when they are connected to the exhaust cavities *12* and *3* which lead to the exhaust steam line.

The outer passages *10*, *9*, *1* and *7* admit live steam into the cylinders.

Valve gear operation is shown schematically in Fig. 20.

In the position of the valves shown in Fig. 20*a* live steam enters the lower end of the left cylinder through the inlet port of passage *9*. Spent steam is exhausted from the upper end of this cylinder through passage *11* and cavity *12*. Under the pressure of the live steam piston *16* with its rod *20* moves upwards pulling the left end of rocker arm *25-27*. Pivoting about its axis *26*, the rocker arm pulls down connecting link *19* and valve rod *28* together with valve *5* of the right cylinder. The rod is linked to the valve by means of block *4* mounted on the threaded part of the valve rod between the two lugs of the valve. Valve gear adjustments can be made by screwing the block along the valve rod.

At the moment observed, valve *5* has just begun its down stroke (steam exhaust passage *2* is closed) and has opened the steam inlet

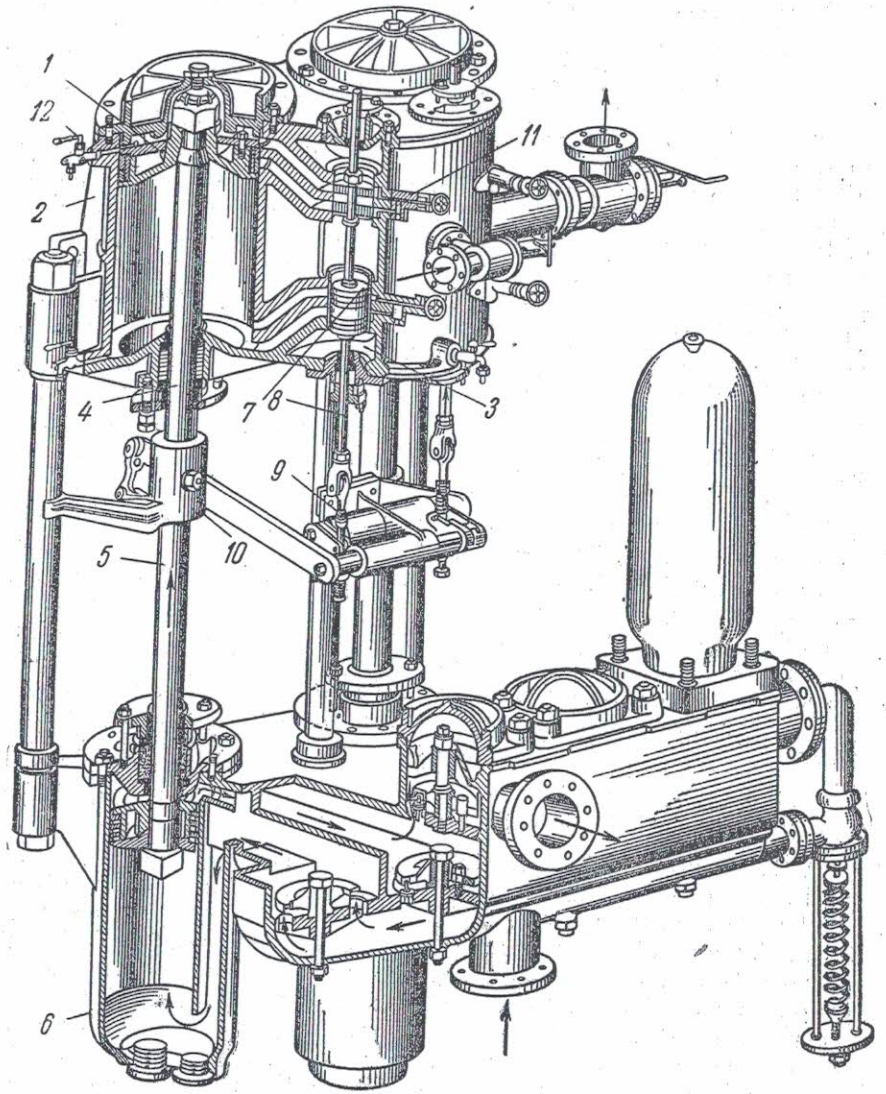


Fig. 19.

port of passage *1* leading to the upper end of the right cylinder. Steam entering the cylinder moves piston *17* downwards and the spent steam in the lower end is exhausted through passage *6* and cavity *3*. Rocker arm *24-22* transmits this downwards movement to connecting link *18*, valve rod *29* and block *14* of the left cylinder.

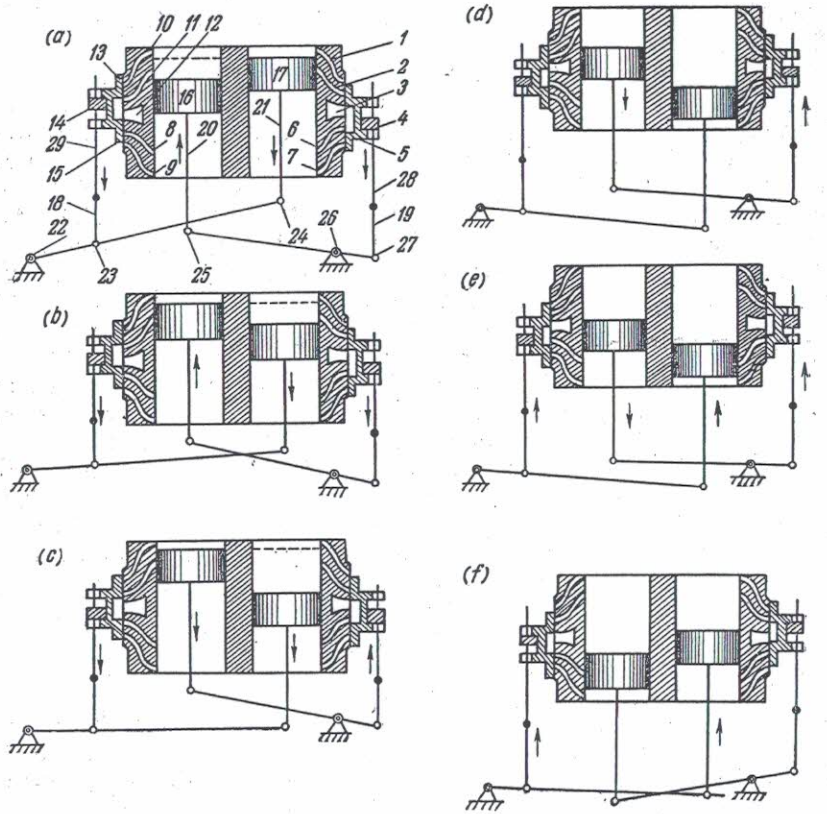


Fig. 20.

However, notwithstanding the movement of block *14*, valve *13* remains stationary until the block reaches the lower lug of the valve, after which it begins to travel downwards too (Fig. 20*b*). This lost motion of the valve enables piston *16* to trap the steam in the upper end of the cylinder after closing the inner port of passage *11*, providing a cushion, and then to come to a stop. Valve *5* also comes to a stop at the end of its down stroke, having fully opened the steam inlet port of passage *1* to the upper end of the right cylinder and

the steam exhaust port of passage *6* from the lower end. At this moment, the piston of this cylinder begins to travel at constant speed.

Upon further downwards motion of piston *17*, valve *13* of the left cylinder begins to travel in the same direction. Piston *16* will remain stationary, however, as long as valve *13* closes the ports of inlet passage *10* to the upper end of the cylinder and exhaust passage *8* from the lower end. The last moment of this position is shown in Fig. 20*c*.

These ports are opened in further downwards travel of piston *17* and, consequently, of valve *13*; piston *16* begins its down stroke pushing block *4* of valve *5* on the right cylinder upwards. While block *4* travels upwards taking up the lost motion, piston *17* closes the inner port of exhaust passage *6*, compresses the trapped steam in the lower end of the cylinder and comes to a stop (Fig. 20*d*). Piston *17* remains in its lower position until the port of lower inlet passage *7* and the port of upper exhaust passage *1* of the right cylinder are opened by the upwards travel of valve *5*. Then piston *17* begins its up stroke. The rest of the cycle (Fig. 20*e* and *f*) proceeds with similar actions of the valve gear.

It is evident from the preceding description that the steam end of the pump operates without lead in steam admission and without steam expansion, and that one piston lags behind the other by one half of a stroke, being in the central position at the same time with the valve it actuates. This latter circumstance is utilized in setting the valves of this type of pump.

Another feature of valve gear operation is that both valves cannot be in their central positions at the same time unless, of course, they are purposely positioned so in setting the valves. This means that the pump can be started from any position of the pistons. The duration of piston dwell at the ends of the stroke and the stroke length depend directly upon the lost motion.

One principle of positive-action pump design is that the piston stroke is reduced if the exhaust ports of the upper and lower ends of a cylinder are brought closer together since this increases the volume of the steam cushion. The stroke can be increased if the ports are designed farther apart.

2. *Valve Setting*

Valve setting, i.e., the adjustment of the valves and valve gear so that the various events of the operating cycle of the steam pump occur at the proper times, depends upon the design of the valve lost-motion mechanism.

Pumps with positive-action valve gear have the following types of adjustment:

(a) inside adjustment with fixed lost motion used for common slide valves,

(b) inside adjustment with adjustable lost motion used for common slide valves, and

(c) outside adjustment with adjustable lost motion used for both common and piston valves.

Inside adjustment with fixed lost motion is employed on pumps in which the amount of lost motion of the valves is set at the manufacturing works. Here, the lost motion mechanism consists of a block with internal thread screwed on the part of the valve rod that is between the lugs of the valve (Fig. 21). The amount of space between the block and lugs constitutes the lost motion.

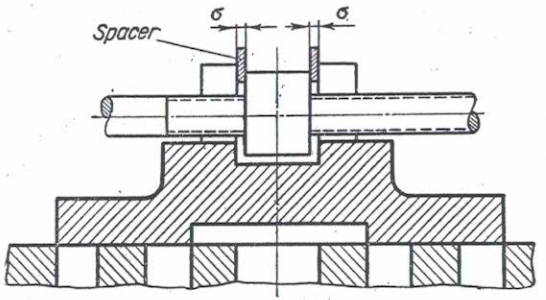


Fig. 21.

The first operation in valve setting is to set the piston of one of the steam cylinders into its central position. This is accomplished by moving it to one extreme position up to the cylinder head and scribing a mark on the piston rod at the face of the packing gland. A second mark is scribed on the rod with the piston in the other extreme position. The distance between the two scribe marks is divided in half, and the piston is moved until this halfway mark comes to the face of the stuffing box. After one piston is thus put into its central position (at midstroke) the valve-chest cover is removed from the adjacent cylinder and the valve rod is disconnected from the connecting link by removing the knuckle joint pin. The valve is moved so that its edges coincide with the edges of the outer steam ports. This shows that the valve is in its central position. Holding the valve in this position, the block is also set into the central position by inserting two spacers σ (Fig. 21) of the same thickness between the block and the valve lugs. Then the valve rod is screwed into or out of the block until the holes coincide in the knuckle joint so that the pin can be replaced.

The valve gear of the second steam cylinder is adjusted in the same manner and then the spacers between the blocks and valve lugs are removed.

Both pistons and both valves are now in the central positions and all the steam ports are closed. In normal operation of the pump the valve gear is never in this position since both steam and exhaust ports are covered by the valves. One of the valves must be moved by hand from this position in order to start the pump.

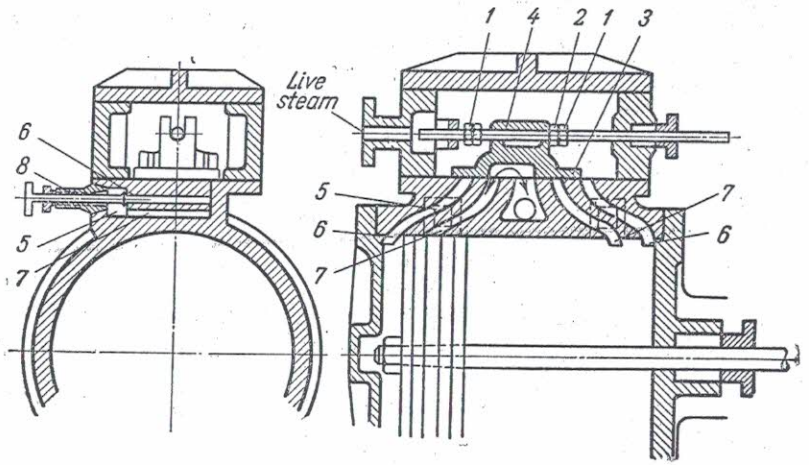


Fig. 22.

To check the valve setting, each valve is moved first in one direction and then in the other, up against the block, with the valve rods and piston rods remaining stationary in the previous central position. The actual steam port opening is measured on each side of the valve. The valve is properly set when equal port openings are obtained.

Piston dwell at the extreme positions depends upon the lost motion, or distances between the block and the valve lugs. Increased distance on any side leads to an increase in the dwell of the piston at the corresponding end of the stroke, and vice versa. The clearance σ usually ranges from 1 to 5 mm.

Inside adjustment with adjustable lost motion involves the following procedure. After removing the chest covers the pistons and valves are set in their central positions in the same manner as in the preceding case. Then, without disconnecting the valve rods from the connecting links, lost-motion nuts *2* are adjusted on each side of each valve lug *4* so that there is a clearance of about half the width of a steam port on each side (Fig. 22). The nuts are first set, valve *3* is moved up against one nut and then up against the other nut, and the actual steam port openings are measured. If the port openings are equal, the lost-motion nuts are locked in position with lock nuts *1*.

The pump is started after replacing the valve-chest covers. It may happen that the piston stroke obtained by the valve setting is too long or too short. In this case, the lost motion is set by a trial-and-error procedure to obtain the required piston stroke. Too much lost motion lengthens the stroke and may cause the pistons to strike the cylinder heads; too short a stroke reduces the pump capacity.

Outside adjustment with adjustable lost motion is widely used in vertical pumps with piston valves. Here the valve is rigidly attached

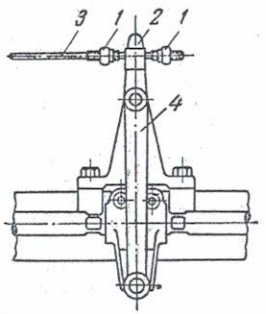


Fig. 23.

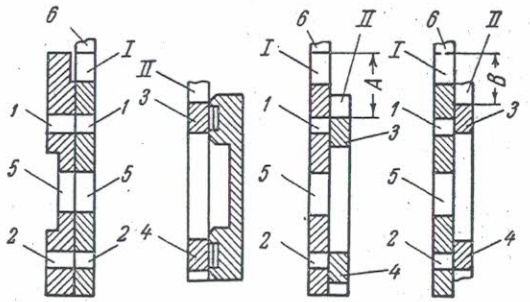


Fig. 24.

to its rod, and the device for adjusting the lost motion is outside of the valve chest (Fig. 23). Adjustment begins by setting the pistons into their central position. Then the valve-chest cover is removed, the valve is set into the central position and lost-motion nuts, or collars, *1*, are set at distances equal to half the steam port width on each side of short rocker arm *2* when long rocker arm *4* is stationary. The adjustment is tested by starting the pump after replacing the valve-chest cover. Too long a piston stroke can be shortened by setting the lost-motion nuts closer together and vice versa.

This lost motion device can also be adjusted without opening the valve chest. This is accomplished by setting the piston into the central position, screwing the lost-motion nuts up against the rocker arm and then screwing them apart gradually and uniformly until the required stroke is obtained. However, if nuts *1* are not at equal distances from the arm or another actuating element when the piston and valve are in their central positions, the piston will travel nearer to one cylinder head than to the other.

In pumps with piston valves valve gear adjustments are made by means of templates. These templates are made of two wooden battens. Marks are made on batten *I* (Fig. 24) coinciding with the edges of ports *1* and *2* of the outer passages and of cavity *5* in the middle of the valve seat, as well as of the upper edge of the valve seat or the edge of the upper flange of the valve-chest stuffing box *6*. The edges of the piston valve collars *3* and *4* are marked on the second batten *II*. Both battens, or templates, are properly marked "Top" and "Bottom" so that they will not be turned over and confused when they are put together. The battens prepared in this manner are used to measure the distances from the mark made on batten *I* (corresponding to the face of the stuffing box or upper edge of the valve seat) to the extreme positions of the valve. These measurements are used in setting the piston valve and securing it on its rod.

In Fig. 24 this distance is denoted as *A* for the lower valve position and as *B* for the upper position. Further adjustments are made as previously described.

These methods of adjustment can provide a definite piston stroke with proper cushioning only for a given condition of pump duty. Under varied conditions the steam pressure in the cushion changes and this affects the piston stroke.

A normal stroke can be maintained at any conditions of load by installing a special valve to regulate the steam pressure in the cushion. Such a cushion valve, of a type used on medium- and high-capacity pumps, is shown in Fig. 22. Two cushion valves *8* are provided for the two ends of the cylinder. They are fitted in chambers *5* which form connections between the corresponding steam *6* and exhaust *7* passages. By opening or closing cushion valve *8* the amount of cushioning steam escaping from the steam passage to the exhaust passage can be regulated and thus the required piston stroke can be obtained for any pump duty.

(B) PUMPS WITH SEMIAUTOMATIC VALVE GEAR

Semiautomatic valve gear is usually employed on simplex pumps which have one steam and one liquid cylinder. The rods of these cylinders are linked together with a crosshead so that their pistons travel simultaneously in the same direction.

In these pumps an auxiliary valve is actuated from the crosshead through suitable linkage. This valve, in turn, controls steam distribution to the main valve which shuttles back and forth admitting live steam to or exhausting spent steam from the steam cylinder of the pump.

A great many designs of valve gear are employed on pumps of this type; some operate with expansion of the steam and some without. Two or three piston or common slide valves are incorporated in these designs. A description follows of the pumps of this type that are most extensively applied in marine installations.

1. Pump with One Piston Valve and One Plain Slide Valve

This type of pump (Figs. 25, 26, 27 and 28) has a plain slide pilot valve *1* and a main piston valve *2* shifted by steam. Live steam is admitted to the chest of pilot valve *1*. This method of steam supply provides for expansion in the steam cylinder of the pump.

The plain slide pilot valve *1* is actuated by rocker arm *3* pivoted on pin *4* in stationary brace *5*, which is secured to valve-chest housing *6* of the pilot valve. The other end of rocker arm *3* passes freely through a slot in pin *7*, mounted freely in crosshead *8*, which links together the rods of the steam and liquid cylinders. Connecting link *9* of valve rod *10* passes freely through tappet *11* of arm *3*. Thus, the pilot valve begins movement only after tappet *11* runs up against lower *13* or upper *14* lost-motion nuts. Live steam is supplied to valve chest *6* of pilot valve *1* through connection *15* and spent steam is exhausted through the opening in flange *16*. During pump operation the steam piston and the pilot valve always travel in the same direction, parallel to the axis of the cylinders. The main piston valve travels in a direction perpendicular to the cylinder axis and to pilot valve travel. The main valve chest has three ports. Left port *17* opens to a passage which leads to the lower end of the steam cylinder; right port *18* leads to the upper end. Port *19* connects to the exhaust steam line. The main piston valve travels in sleeves *20* (Fig. 26) having handles *21* outside the valve chest for turning the sleeves by hand. The sleeves have shaped slots *22* and *23*. The main piston valve is hollow, having three cavities and a flat with five ports on the side farthest away from the steam cylinder. Port *24* is connected to left cavity *25* and port *26* on the cylindrical surface of the valve. Middle cavity *27* is connected to ports *28* and *29* on the cylindrical surface and port *30* on the flat. Port *31* on the flat is connected through right cavity *32* to port *33* on the cylindrical surface of the valve.

Passage *38* connects port *34* on the flat with port *36* on the left end face of the piston valve, while port *35* is connected by passage *39* to port *37* on the right end face.

Pilot valve *1* has a shaped recess *40* on the seat side.

The piston and valve gear of the steam end of the pump interact in the following manner. Let us assume that the steam piston is in its extreme upper position (Fig. 25). The pilot valve will also be in its extreme upper position and the main piston valve in its

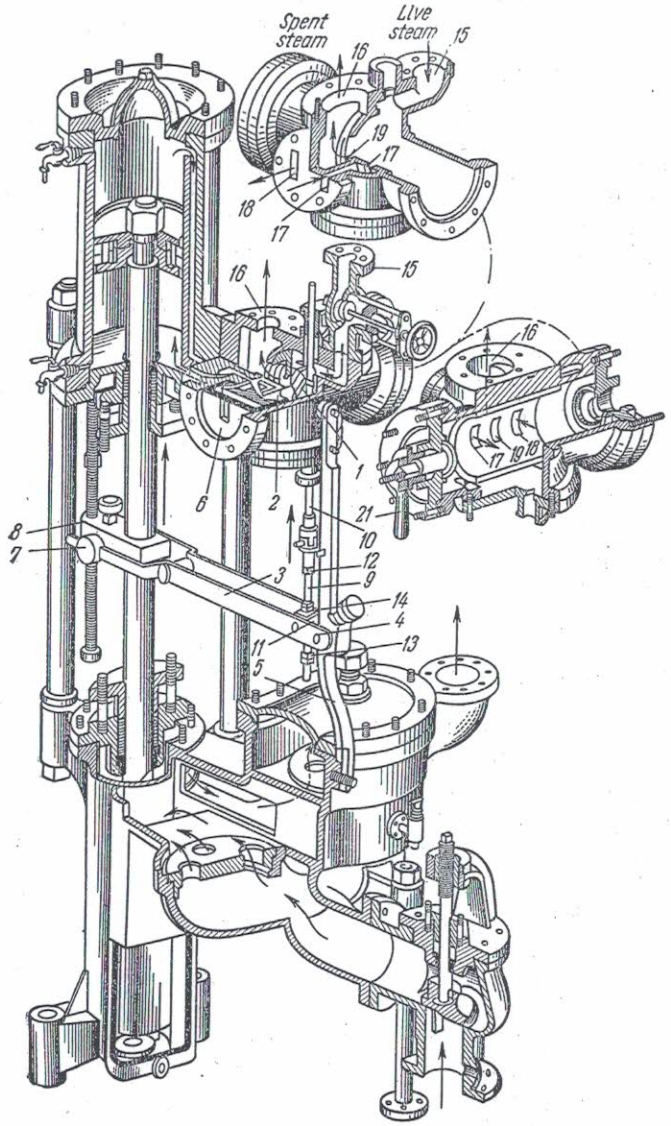

Spent steam
Live steam

Fig. 25.

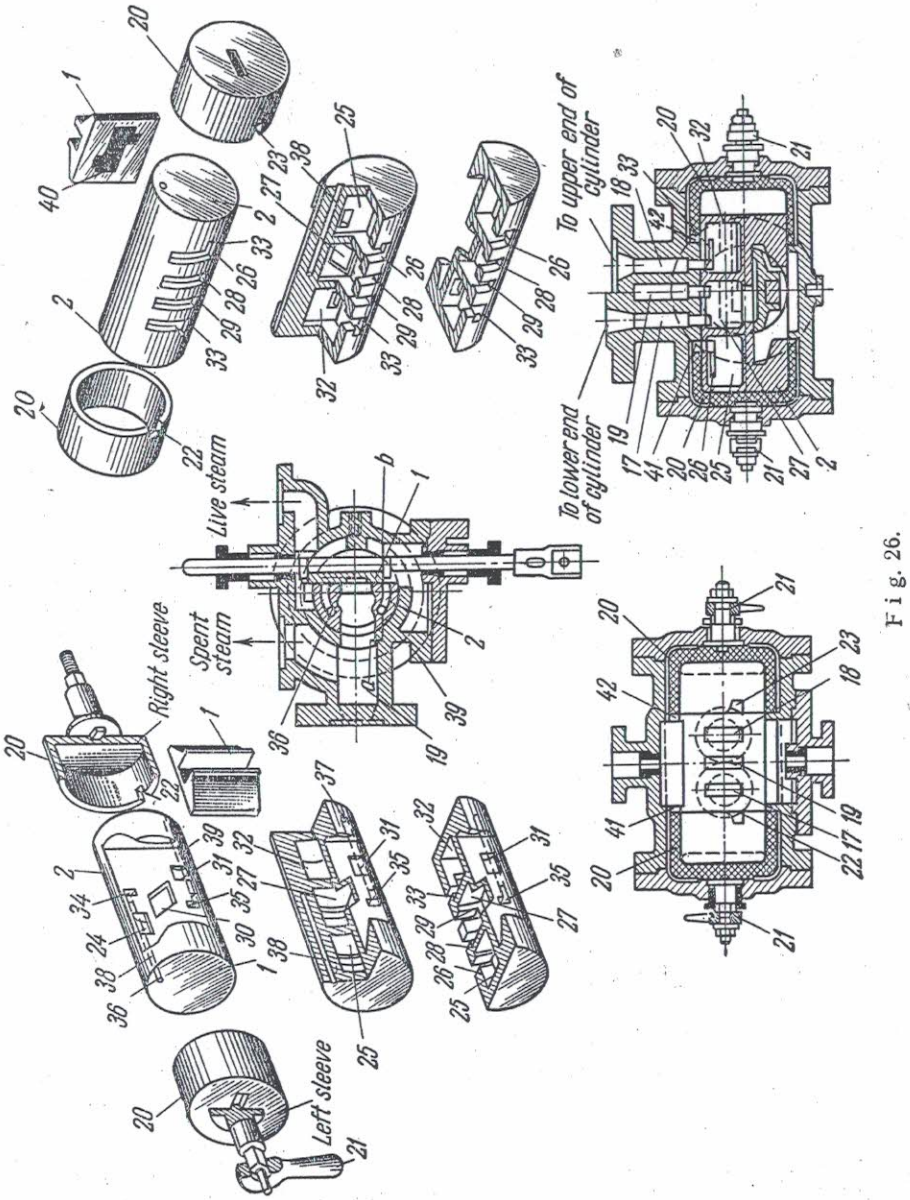

To upper end of cylinder
To lower end of cylinder
Live steam
Spent steam
Right sleeve
Left sleeve

Fig. 26.

extreme left position. The relative positions of the main and pilot valves are shown in Fig. 26.

The arrangement of the main *2* and pilot *1* valves for various positions of the steam piston is shown schematically in Fig. 27.

The main piston valve is in the extreme left position, shown in Figs. 26 and 27*a*, because it is held there by live steam admitted

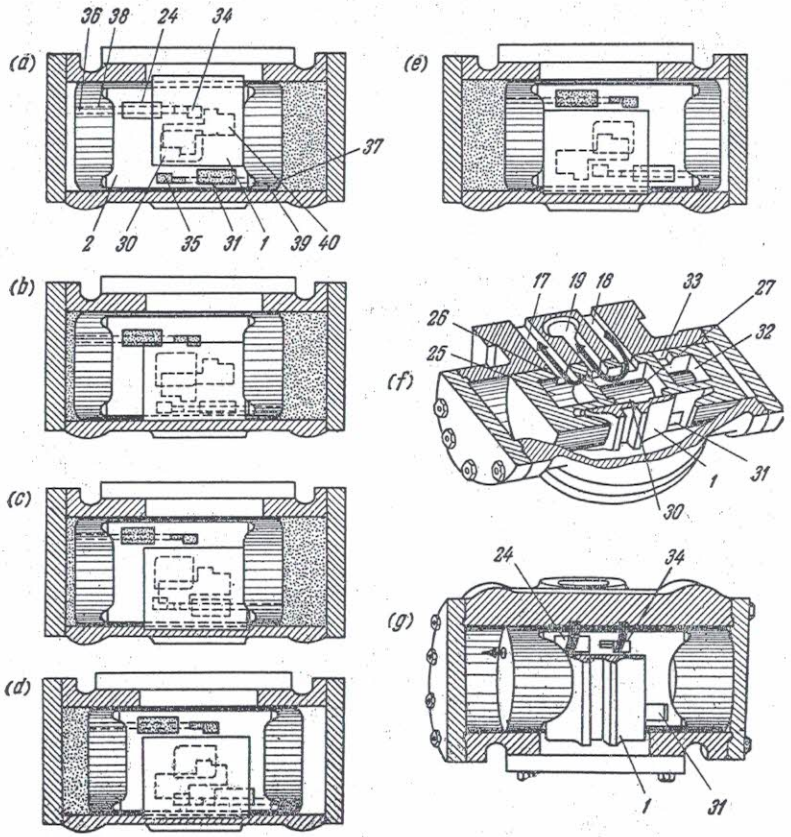


Fig. 27.

through port *35*, passage *39* and port *37* into the right sleeve of the valve. Live steam is also admitted through ports *24* and *31*. Steam passes through port *24*, cavity *25* and port *26* where it is stopped by a blank wall (Fig. 26). The other port *31* admits steam through cavity *32*, port *33* and passage *18* to the upper end of the steam cylinder, forcing the piston downwards. Spent steam is exhausted, at

the same time, from the lower end through passage *17*, port *28*, cavity *27* and passage *19* to the exhaust line.

As the steam piston travels downwards it does not at first actuate the pilot valve. The latter begins to move when tappet *11* of rocker arm *7-4* runs up against nuts *13*. This occurs when the piston has travelled about 0.6 of its stroke. At 0.75 of its stroke the piston puts the pilot in a position where it closes port *31*. In this position ports *31* and *35* are closed so that live steam cut-off occurs. The main piston valve remains in its extreme left position while the piston continues its down stroke under the pressure of the expanding steam pulling rocker arm *7-4* and the pilot valve farther downwards.

In its downward movement the pilot valve opens port *34* (Fig. 27*b*) and live steam is admitted into the left sleeve of the main valve through passage *38* and port *36*. The main valve will not shift to the right, however, until the pilot valve reaches the position shown in Fig. 27*c*, since the steam pressure is equal in the left and right sleeves. In the position of Fig. 27*c* recess *40* of the pilot valve connects port *35* to port *30* of middle cavity *27* which is connected to the steam exhaust line. At this, the steam pressure in the right sleeve drops and the main valve is shifted by the live steam to its extreme right position (Fig. 27*d*). Just before it reaches this position, port *35* moves away from recess *40* in the pilot valve, thereby cutting off steam exhaust from the right sleeve. The steam remaining in the sleeve is compressed by the moving main valve, forming a cushion and stopping the valve before it can strike the bottom of the sleeve (Fig. 27*e*).

As the main valve travels to the right (Fig. 27*d*) first passages *17* and *19* are cut off from cavity *27*. This reduces steam exhaust from the lower end of the steam cylinder, the piston compresses the remaining steam and comes to a smooth stop. This compression of the steam continues until cavity *25* becomes connected to passage *17* through ports *24* and *26*, thereby admitting live steam into the lower end of the steam cylinder (Fig. 27*e*, *f* and *g*). Since at this time the upper end of the cylinder is connected with the exhaust line through passage *18*, ports *28* and *29* and the middle cavity *27*, the piston of the steam cylinder starts its up stroke. If the pump is stopped in a position where the pilot valve closes the steam admission ports it cannot be subsequently started by merely turning on the steam. In such cases it will be necessary to turn the sleeves by their handles *21* so that shaped slots *22* and *23* on the sleeves register with ports *26* and *33*.

Let us assume that the pump must be started with the pilot valve in the position shown in Fig. 26. This can be done if the right sleeve is turned so that its slot *23* registers with port *33* of the main valve. This admits live steam through slit *41* (Fig. 26) into the space

between the sleeves and the valve chest, from where it passes through slit *42*, slot *23* and the angular end of port *33* to cavity *32*, and further to the upper end of the steam cylinder.

The valve gear may be adjusted to obtain the required piston stroke and point in the stroke where live steam cut-off occurs. Such adjustments are made by means of lost-motion nuts *13* and *14*, sleeve nut *12* and connecting link *9* (Fig. 25). Moving the nuts closer together shortens the stroke and, consequently, the pump capacity. Moving them apart lengthens the stroke and increases the capacity. The stroke should never be lengthened to the point where the piston strikes the cylinder heads at the ends of the stroke.

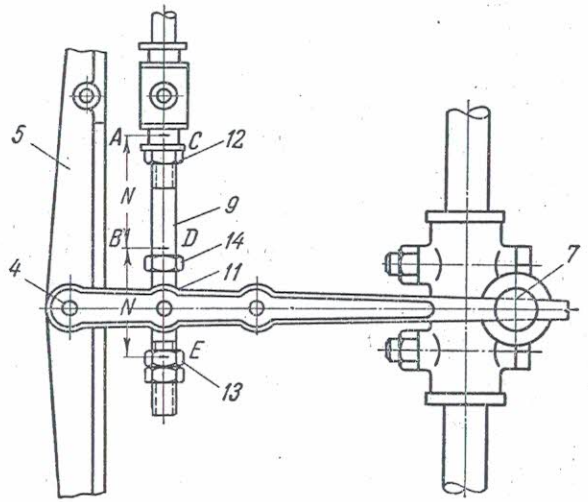


Fig. 28.

The valve gear may be checked and set according to setting marks, if such have been made in the manufacturing works during pump assembly. As a rule, centrepunch marks *A* and *B* are provided on brace *5* (Fig. 28), mark *C* on the sleeve nut, mark *D* on the connecting link and mark *E* on nut *13*. In designs in which nut *14* is fastened rigidly on the link, the distance *N* between marks *C* and *D* is equal to the distance between marks *A* and *B*. This is easily achieved by screwing connecting link *9* into or out of the sleeve nut.

Then nut *13* is adjusted so that the distance *N* between marks *D* and *E* is equal to the distance *AB*. No further adjustments are required when works' marks are employed for valve setting. If nut *14* is adjustable it will carry mark *D*. In the absence of setting marks, the piston is set into the position where steam cut-off occurs and the

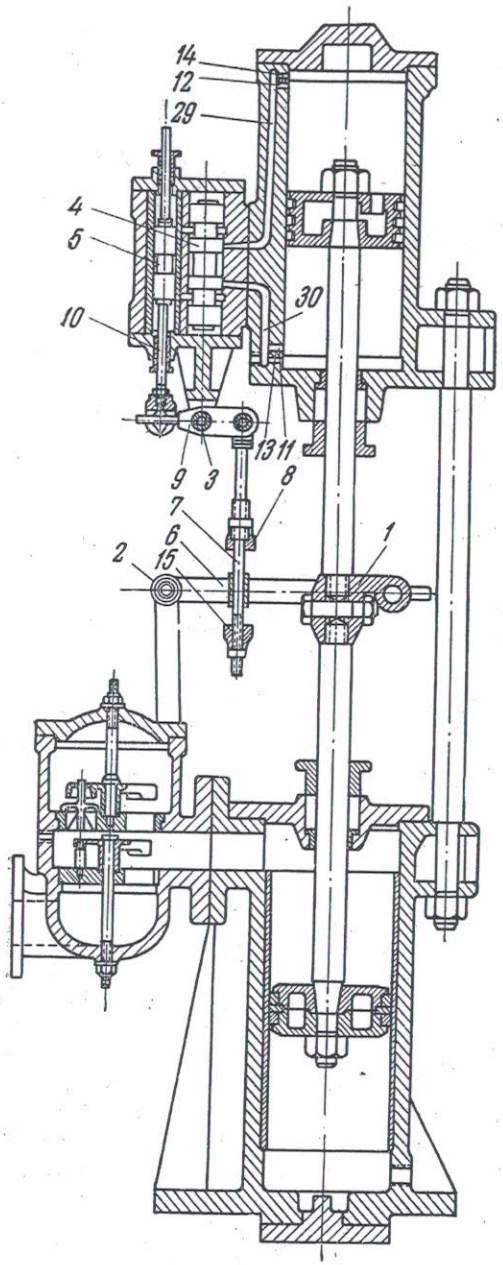


Fig. 29.

pilot valve is set in the corresponding position. Tappet *11* must be up against nut *13* on the down stroke of the piston or against nut *14* on the up stroke. The main piston valve should, of course, be in the corresponding end position when these adjustments are made.

2. *Pump with Two Vertical Piston Valves*

The valve gear of these pumps (Figs. 29 and 30) consists of main rocker arm *6* pivoted at *2*, connecting link *7* with lost-motion nuts *15* and *8*, auxiliary rocker arm *9* pivoted at *3* and jointed to valve rod *10*, auxiliary piston valve *5* and main piston valve *4*.

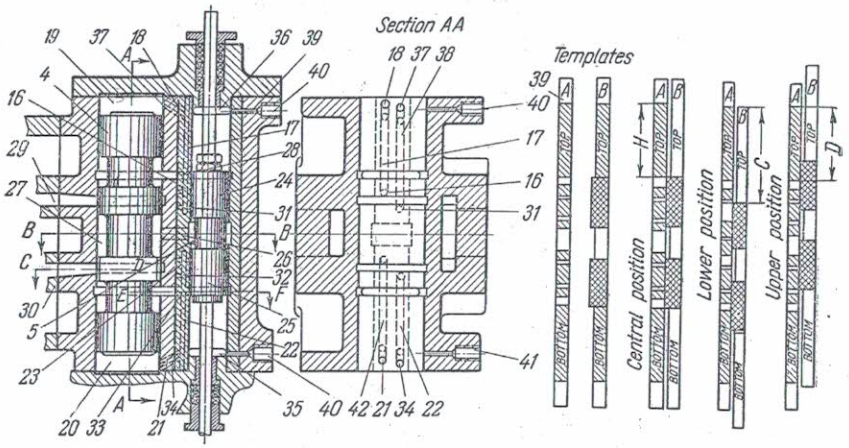


Fig. 30

Upon travel of the pistons, crosshead *1* actuates cross arm *6*, and the latter shifts auxiliary valve *5* in the direction opposite to pump piston travel.

Let us follow through the operation of the valve gear assuming that the steam piston begins its stroke from the extreme upper position. Here, auxiliary valve *5* is in its lower position and cross arm *6* is up against nut *8*.

Live steam, admitted through port *36* (Fig. 30) into the upper end of the chest of auxiliary valve *5*, enters the upper end of the main valve chest through passages *16*, *17* and *18*. At the same time, lower end *20* of the main valve chest is connected through passages *21*, *22* and *23* to the recess between pistons *24* and *25* of the auxiliary valve and further through passage *26* to exhaust steam cavity *27*. The difference in pressure on the top and bottom of the main valve shifts it to its lower position.

Live steam is admitted through passages *28* and *29* and port *14* into the upper end of the steam cylinder, thus moving the piston downwards. Spent steam from the lower end of the cylinder is exhausted through ports *11* and *13*, passage *30* and cavity *27*.

When the piston has travelled downwards a distance equal to 0.6 of its full stroke, cross arm *6* runs up against lost-motion nut *15* and moves the auxiliary valve upwards. This valve simultaneously closes the ports of passages *16* and *23*; this occurs when the steam piston has travelled 0.75 of its stroke. In its subsequent upward movement the auxiliary valve closes the port of passage *28* thereby cutting off the admission of live steam to the upper end of the steam cylinder and its piston continues its stroke under the action of the expanding steam. As the piston approaches its lower position the edge of piston *25* on the auxiliary valve first opens the port of passage *33* and then that of passage *32*. Moreover, the lower edge of piston *24* on this valve opens the port of passage *31*.

Now, live steam, supplied through port *35*, is admitted through passages *32* and *34* into lower end *20* of main valve *4*, while its upper end *19* will communicate through passages *37*, *38*, *31* and *26* with the exhaust steam cavity *27*. The live steam shifts main valve *4* upwards.

As it approaches its upper position, the upper piston of the main valve closes the port of passage *18* and the valve is stopped by the cushion formed in upper end *19* of the main valve chest. In this position, live steam from the lower end of the auxiliary valve is admitted through passages *33* and *30* and port *11* into the lower end of the pump steam cylinder, moving the piston upwards. As soon as port *13* is opened it also begins to admit live steam.

At the beginning of the stroke spent steam is exhausted from the upper end of the pump cylinder through port *14* and passage *29* into the exhaust steam cavity *27*. Spent steam also passes through port *12* as soon as it is opened by the piston. Blow-down cocks for the main and auxiliary valve chests are mounted in threaded holes *40* and *41*.

Since live steam is admitted to both the upper and lower ends of the piston valve, the latter is completely balanced. As the steam piston approaches the upper and lower ends of its stroke it closes ports *12* and *11*, respectively, forming a steam cushion that slows down and smoothly stops the piston in its extreme positions.

The lost-motion mechanism is adjusted by means of nuts *8* and *15* on connecting link *7*.

Templates are employed to adjust the valve gear of such pumps; they enable all the moments in steam distribution to be followed through.

The positions of the ports in the seat of the auxiliary valve chest are marked on template *A*; the piston edges of the auxiliary valve and upper face *39* of the chest are marked on template *B*. The auxiliary valve should be in its central position when the steam piston has travelled a distance of 0.7 to 0.75 of its stroke from the upper or lower position.

After putting templates *A* and *B* together so that the marking representing the ports *33* and *28* are covered by the marking of the valve pistons *25* and *24*, we measure the distance *H*. This is the required distance between the top edge of valve piston *24* and upper face *39* of the valve chest when the auxiliary valve is in its central position.

Next, the valve chest head is removed and the upper face of piston *24* of the auxiliary valve is positioned at the distance *H* from upper chest face *39*.

The piston of the steam cylinder is set to a position from 0.7 to 0.75 of its stroke as measured from the upper extreme position, and lower lost-motion nut *15* is moved up against rocker arm *6*. Upper lost-motion nut *8* is adjusted in the same way after setting the steam piston at the same distance as before, but from the extreme lower position. Then the steam piston is shifted to its extreme upper position and the distance *C* is measured. It is measured between the same elements as the distance *H* but in a new position of the auxiliary valve and should equal the distance *C* as measured on the templates when correspondingly positioned. This procedure is repeated for the extreme lower position of the steam piston, measuring the distance *D*.

Finally, the upper and lower positions of the steam piston are scribed on the rod, the valve chest head is replaced and the pump is started for a test run.

If the piston stroke is too short the lost-motion nuts should be screwed apart, if it is too long they should be screwed towards each other until the required stroke is obtained.

3. Pump with One Vertical and One Horizontal Piston Valve

Two piston valves comprise the valve gear in this type of pump (Fig. 31). Auxiliary valve *3* and its rod *7* are actuated through a connecting link from a rocker arm oscillated by a crosshead on the pump piston rods. Consequently, the steam piston and auxiliary valve *3* travel vertically in different directions during pump operation.

Main piston valve *16* is actuated by the pressure of steam admitted to and exhausted from the main valve chest by auxiliary valve *3* having pistons *2* and *5*. Main valve *16* travels horizontally. Live

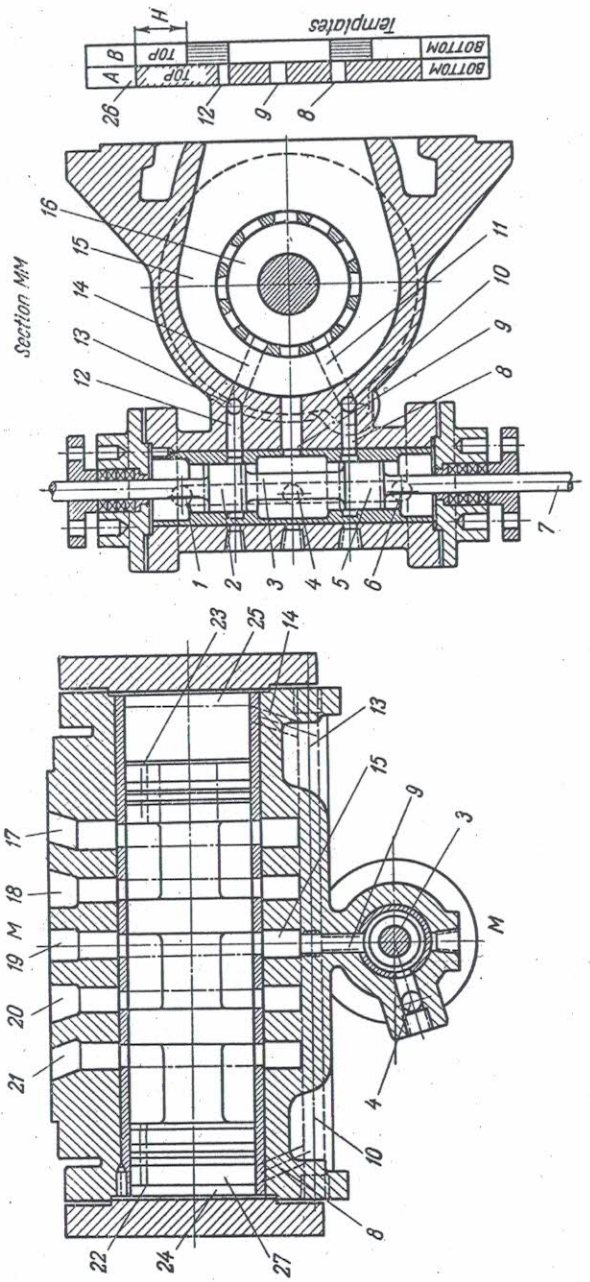


Fig. 31.

steam is admitted into the chest through passages *17* and *21*. Passage *18* is connected to the upper end of the steam cylinder, passage *20* to the lower end, and passage *19* to the steam exhaust line.

In the position shown in the figure live steam enters through passage *21* and passage *22* in piston *27* of the main valve to left end *24* of the main valve chest where it is cut off by the edge of the main valve. Simultaneously live steam is admitted through passages *17*, *23*, *14* and *13* to the port of passage *12* which is closed by the edge of piston *2* on auxiliary valve *3*. Main valve *16* remains stationary since the pressure is equal at its ends.

Live steam is also admitted into the upper end of the steam cylinder through passages *17* and *18* and it pushes the piston downwards. By means of the crosshead, rocker arm, connecting link and rod *7*, the auxiliary valve is pushed upwards. Spent steam from the lower end of the steam cylinder leaves through passages *20* and *19* which are connected to the steam exhaust line. When the piston reaches its lower position, the auxiliary valve will be in its upper position in which it opens the port of passage *12* and connects right end *25* of the main valve chest, through passages *14*, *13*, *12* and *9*, with steam exhaust cavity *15*. The pressure of the live steam in the left end *24* of the main valve chest will shift the main valve to its right position. Before it reaches this position, however, it closes the port of passage *14*, thereby trapping steam and forming a cushion which brings it to a stop. Now live steam enters the lower end of the steam cylinder through passages *21* and *20* and spent steam from the upper end is released through passages *18* and *19* to the exhaust line. This starts the steam piston on its up stroke. Auxiliary valve *3* begins moving when the rocker arm reaches the lost-motion nut on the connecting link. The piston will have travelled a certain part of its stroke by this time.

Thus actuated, auxiliary valve *3* shifts to its lower position where it connects the left end *24* of the main valve chest, through passages *11*, *10*, *8* and *9*, with the exhaust cavity. The pressure of live steam entering through passage *23* shuttles the main valve to its extreme left position and the down stroke of the steam piston begins.

Ports *1*, *4*, *6* and their passages connect the upper and lower ends of the auxiliary valve chest, by means of port *4* and passage *9*, with exhaust steam cavity *15*. This feature helps to balance the auxiliary valve and relieves the stuffing boxes of its rods of high pressures thereby simplifying the design of the stuffing boxes and reducing steam leakage through the packing.

The valve gear is adjusted with the aid of templates *A* and *B* marked off, one for the seat of the auxiliary valve and the other for the valve itself. The templates are put together in the position

representing the central position of the valve in its chest. Then distance *H*, between mark *26* (representing the upper face of the valve chest) on template *A* and the mark on template *B* indicating the upper end of the valve, is measured. Based on this measured distance the auxiliary valve is set into its central position. Then the steam piston is set into its central position and the upper and lower lost-motion nuts are adjusted on each side of the cross arm leaving clearances equal to one half of the steam port width on each side. The auxiliary valve is moved up or down so that one of the nuts contacts the cross arm as otherwise the pump cannot be started. Finally, the valve gear is reassembled and the pump is started for a test run in which the length of stroke is obtained by further adjustment of the lost-motion nuts.

(C) PUMPS WITH AUTOMATIC VALVE GEAR

These pumps differ from the single-acting simplex pumps described previously only in that they do not have a valve gear drive and auxiliary valves. The well-known Cameron pumps are of this type.

1-7. Determining the Head of Liquid Acting on the Piston End During the Suction and Discharge Strokes

(A) SUCTION STROKE OF THE PISTON

Normal pump operation requires that the liquid in the cylinder follows directly after the piston (or plunger) forming a continuous flow from the inlet strainer to the end of the piston.

The variation in the total energy (head) of the flow of liquid at any point in the suction pipeline may be expressed by the equation for single-dimensional unsteady motion

$$\frac{\partial}{\partial l}\left(z+\frac{p}{\gamma}+\frac{v^2}{2g}\right)+\frac{1}{g}\frac{\partial v}{\partial t}+\frac{\partial \zeta}{\partial l}=0 \qquad (43)$$

The first term in this equation is a derivative of the total specific energy per kilogram of liquid with respect to the distance l, i.e., the variation in this energy (head) per unit of distance travelled.

The second term expresses the variation in energy (head) of the inertia forces of one kilogram of liquid per unit of length travelled depending upon the law governing the acceleration $\frac{\partial v}{\partial t}$ of the liquid.

The last term represents the specific energy (head) required to overcome hydraulic resistances in the pipeline per unit of length

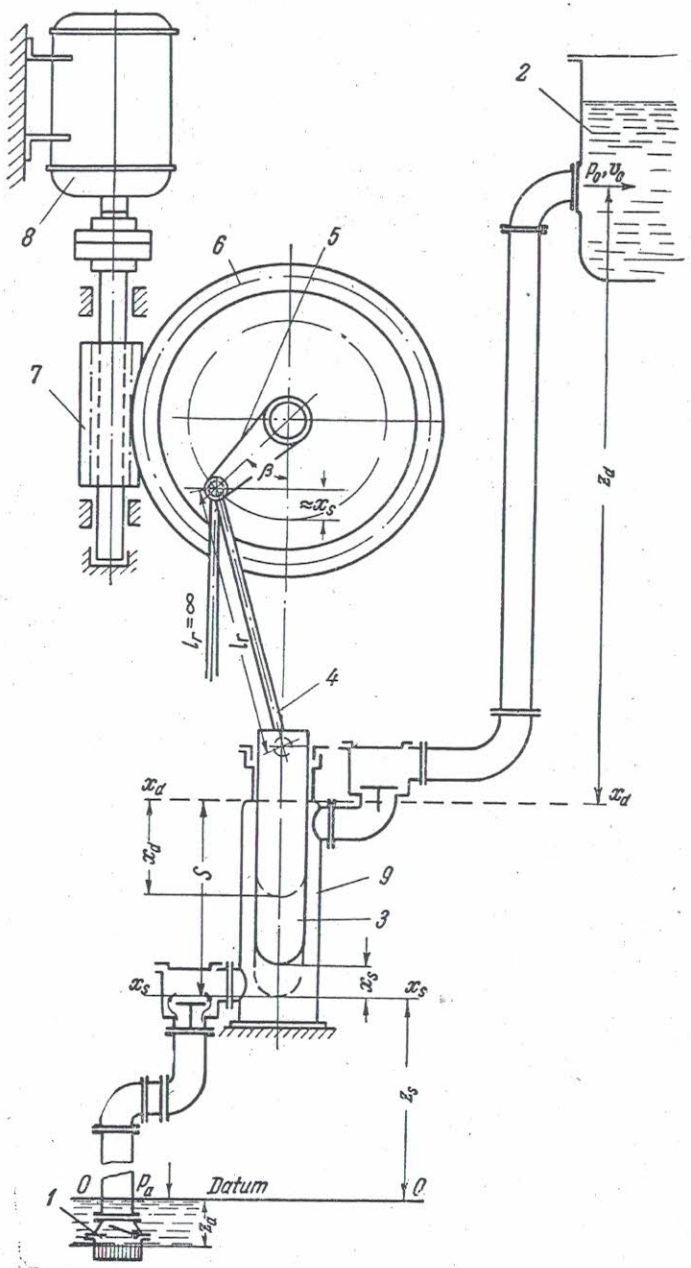


Fig. 32.

travelled by the liquid. This term is not a derivative, however, since the hydraulic resistance is the sum of the friction resistance along the pipeline and local resistances which are not a continuous function of the distance travelled l. It follows that the last term is written as a derivative only to preserve the homogeneous form of the equation. We can now proceed to find the law by which the head on the end of the piston varies during the suction stroke of the pump. This can be done on the example of the vertical motor-driven power plunger pump shown schematically in Fig. 32.

Plunger *3* in cylinder *9* is reciprocated by connecting rod *4* which is linked by crankpin *5* to worm wheel *6*. The latter is driven through worm *7* from the shaft of motor *8*.

During the up stroke of plunger *3*, liquid is drawn into cylinder *3* through strainer *1*; during the down stroke the liquid is discharged into tank *2*.

Let us use the following notation:

p_a = pressure on the surface of liquid level, kg per sq m
p_s = variable pressure exerted by the liquid on the plunger during the suction stroke, kg per sq m
h_s = loss in head to overcome hydraulic resistances in the inlet (foot) valve, m
$x_s x_s$ = extreme lower position of the plunger
x_s = instantaneous distance from the plunger to its lower position during the suction stroke, m
l_s = length of travel of the liquid from the lower edge of the strainer to the lower end of the plunger in any position of the latter, m
S = plunger stroke, m
z_s = distance between the lower end of the plunger in its lower position and the level of the liquid being pumped from the lower tank, m
z_a = depth of immersion of the inlet holes in the lower strainer below the surface of liquid level, m
c = variable velocity of the plunger, m per sec
t = interval of time, sec
v_s = variable velocity of liquid flow in the suction pipe, m per sec
F = area of the end face of the plunger, sq m
D = diameter of the plunger, m
f_{is} = area of the clear opening in the suction pipe at various points, sq m
d_{is} = inside diameter of the suction pipe at various points, m
λ_{is} = coefficient of friction losses upon movement of the liquid at various points in the pipeline
ζ_{is} = coefficient of local resistances.

The surface of liquid level is taken as the datum.

The equation of unsteady motion will take the following form for dl_s, an infinitesimal length of travel of the liquid being pumped:

$$\frac{\partial}{\partial l_s}\left(z+\frac{p}{\gamma}+\frac{v_s^2}{2g}\right)dl_s+\frac{1}{g}\frac{\partial v_s}{\partial t}dl_s+\frac{\partial \zeta_s}{\partial l_s}dl_s=0 \qquad (44)$$

Integrating this expression in respect to the length of travel of the liquid from the inlet holes of the strainer to the lower end of the plunger in its upper position, we obtain the variation in energy (head) of the moving liquid during the suction stroke of the plunger:

$$\int_{l_s}\frac{\partial}{\partial l_s}\left(z+\frac{p}{\gamma}+\frac{v_s^2}{2g}\right)dl_s+\frac{1}{g}\int_{l_s}\frac{\partial v_s}{\partial t}dl_s+\sum_{l_s}\frac{\partial \zeta_s}{\partial l_s}dl_s=0$$

The summation sign is used before the last term since the loss of energy (head) expressed by the term is not a continuous function along the pipeline.

Using the possibility of integrating the first term by parts and the occurrence of a total differential in the expressions under the integrals, as well as the limits of integration, we can write

$$\int_{l_s}\frac{\partial}{\partial l_s}\left(z+\frac{p}{\gamma}+\frac{v_s^2}{2g}\right)dl_s=\int_{l_s}\frac{\partial}{\partial l_s}(z)\,dl_s+$$

$$+\int_{l_s}\frac{\partial}{\partial l_s}\left(\frac{p}{\gamma}\right)dl_s+\int_{l_s}\frac{\partial}{\partial l_s}\left(\frac{v_s^2}{2g}\right)dl_s=$$

$$=\int_{-z_a}^{z_s+x_s}dz+\frac{1}{\gamma}\int_{p_a+z_a\gamma}^{p_s}dp+\frac{1}{2g}\int_0^c dv_s^2=(z_s+x_c)+$$

$$+z_a+\frac{1}{\gamma}\left[p_s-(p_a+z_a\gamma)\right]+\frac{1}{2g}c^2=z_s+$$

$$+x_s+\frac{p_s}{\gamma}-\frac{p_a}{\gamma}+\frac{c^2}{2g}$$

The expression of the second term of equation (44) under the integral is not a total differential of variable integration and, moreover, we do not know the law by which the acceleration of the liquid varies along its whole length of travel. The second term of equation (44) can be represented as the sum of the variations in the inertia head (energy) over the separate component lengths of the pipeline.

Applying the equation of continuity $f_{is}v_{is}=Fc$ for each section of the suction line with the same diameter we can write

$$v_{is}=\frac{F}{f_{is}}c \qquad (45)$$

then

$$\frac{\partial v_{is}}{\partial t}=\frac{F}{f_{is}}\frac{\partial c}{\partial t} \qquad (46)$$

Taking into account that $\frac{\partial c}{\partial t}$ does not depend upon the limits of integration, and dividing the length travelled by the liquid from the strainer to the lower end of the plunger into lengths l_{1s}, l_{2s},... $l_{xs}=x_sc$ with areas of the clear opening of f_{1s}, f_{2s}, ... $f_{xs}=F$, respectively, then

$$\frac{1}{g}\int_{l_s}\frac{\partial v_s}{\partial t}dl_s=\sum_{l_s}\int_{l_{is}}\frac{1}{g}\frac{\partial v_{is}}{\partial t}dl_{is}=$$

$$=\frac{1}{g}\sum_{l_s}\int_{l_{is}}\frac{F}{f_{is}}\frac{\partial c}{\partial t}dl_{is}=\frac{\partial c}{\partial t}\left(\frac{1}{g}\int_{l_{1s}}\frac{F}{f_{1s}}dl_{1s}+\right.$$

$$\left.+\frac{1}{g}\int_{l_{2s}}\frac{F}{f_{2s}}dl_{2s}+\ldots.+\frac{1}{g}\int_{l_{xs}=x_s}\frac{F}{F}dl_{xs}\right)=$$

$$=\frac{1}{g}\left(\frac{F}{f_{1s}}l_{1s}+\frac{F}{f_{2s}}l_{2s}+\ldots..+x_s\right)\frac{\partial c}{\partial t}=$$

$$\frac{1}{g}\left(\frac{D^2}{d_{1s}^2}l_{1s}+\frac{D^2}{d_{2s}^2}l_{2s}+\ldots.+x_s\right)\frac{\partial c}{\partial t}=\frac{L_s+x_s}{g}\frac{\partial c}{\partial t}$$

In the expression thus obtained the sum of the terms of the type $\frac{F}{f_{is}}l_{is}$ or $\frac{D^2}{d_{is}^2}l_{is}$, having dimensions of length and referring to the part of the pipeline from the strainer to the cross section x_sx_s, is denoted as L_s and is called the referred length of the suction line.

The last addend of equation (44) consists of the losses of head in overcoming friction resistance and local resistances to flow along the length of the suction line, including the pump cylinder.

Specific features of the variation in the loss of head in the foot valve, both in magnitude and the law by which it is governed, make it more expedient to specify these losses as a separate term h_s. Then

$$\sum_{l_s}\frac{\partial \zeta_s}{\partial l_s}dl_s=\sum_{l_s}\lambda_{is}\frac{l_{is}}{d_{is}}\frac{v_{is}^2}{2g}+\sum_{l_s}\zeta_{is}\frac{v_{is}^2}{2g}+h_s$$

Expressing the velocity of the liquid v_{is} at various points in the suction line in terms of plunger velocity, according to equation (45)

we can write

$$\sum_{l_s}\frac{\partial\zeta_s}{\partial l_s}\,dl_s=\sum_{l_s}\lambda_{is}\frac{l_{is}}{d_{is}}\left(\frac{F}{f_{is}}c\right)^2\frac{1}{2g}+\sum_{l_s}\zeta_{is}\left(\frac{F}{f_{is}}c\right)^2\frac{1}{2g}+$$

$$+h_s=\frac{c^2}{2g}\left[\sum_{l_s}\lambda_{is}\frac{l_{is}}{d_{is}}\left(\frac{F}{f_{is}}\right)^2+\sum_{l_s}\zeta_{is}\left(\frac{F}{f_{is}}\right)^2\right]+$$

$$+h_s=\frac{c^2}{2g}\left[\sum_{l_s}\lambda_{is}\frac{l_{is}}{d_{is}}\left(\frac{D}{d_{is}}\right)^4+\sum_{l_s}\zeta_{is}\left(\frac{D}{d_{is}}\right)^4\right]+h_s=$$

$$=w_s\frac{c^2}{2g}+h_s$$

The sum within the brackets of this equation is denoted as w_s and is called the sum of the referred coefficients of resistance of the suction line.

Substituting the results of integration into equation (44) we obtain

$$z_s+x_s+\frac{p_s}{\gamma}-\frac{p_a}{\gamma}+\frac{c^2}{2g}+\frac{L_s+x_s}{g}\quad\frac{\partial c}{\partial t}+w_s\frac{c^2}{2g}+h_s=0.$$

Therefore, the pressure head of liquid acting on the plunger during the suction stroke at any plunger position is equal to

$$\frac{p_s}{\gamma}=\frac{p_a}{\gamma}-\left[(z_s+x_s)+(1+w_s)\frac{c^2}{2g}+h_s+\frac{L_s+x_s}{g}\quad\frac{\partial c}{\partial t}\right]\qquad(47)$$

This formula can be more conveniently applied in actual dynamic lift calculations by expressing the velocity of the plunger and its travel as a function of the angle of crankpin rotation $\beta°$.

The length of travel of a plunger from its lower position in a crank mechanism with a connecting rod of infinite length and a crankpin radius of r is:

$$x_s=r-r\cos\beta°=r(1-\cos\beta°)$$

then

$$\cos\beta°=\frac{r-x_s}{r}=1-\frac{x_s}{r}$$

The velocity, c, square of the velocity, c^2, and the acceleration, $\frac{\partial c}{\partial t}$, of the plunger are:

$$c=\frac{\partial x_s}{\partial t}=\frac{\partial}{\partial t}\,[r(1-\cos\beta°)]=r\sin\beta°\frac{\partial\beta°}{\partial t}=r\omega\sin\beta°$$

$$c^2=r^2\omega^2\sin^2\beta°=r^2\omega^2(1-\cos^2\beta°)=r^2\omega^2\left[1-\left(1-\frac{x_s}{r}\right)^2\right]=$$

$$=r^2\omega^2\left(1-1+2\frac{x_s}{r}-\frac{x_s^2}{r^2}\right)=r^2\omega^2\left(\frac{2x_s}{r}-\frac{x_s^2}{r^2}\right)$$

$$\frac{\partial c}{\partial t}=\frac{\partial}{\partial t}(r\omega\sin\beta°)=r\omega\cos\beta°\frac{\partial\beta°}{\partial t}=r\omega^2\cos\beta°=r\omega^2\left(1-\frac{x_s}{r}\right)$$

The angular velocity ω of the pump shaft at a crankpin speed of n rpm is

$$\omega=\frac{2\pi n}{60}$$

Substituting these values of the velocity and acceleration into equation (47)

$$\frac{p_s}{\gamma}=\frac{p_a}{\gamma}-\left[(z_s+x_s)+(1+w_s)\frac{r^2\omega^2}{2g}\left(\frac{2x_s}{r}-\frac{x_s^2}{r^2}\right)+\right.$$

$$\left.+h_s+\frac{L_s}{g}r\omega^2\left(1-\frac{x_s}{r}\right)+\frac{x_s}{g}r\omega^2\left(1-\frac{x_s}{r}\right)\right]\qquad(48)$$

Each term of formula (48) can be plotted graphically against the various positions of the plunger, thereby enabling its effect to be estimated on the final value of the pressure under the plunger (dynamic lift) at various positions in the suction stroke. Here, the travel of the plunger is plotted along the axis of abscissas and the corresponding values of the terms in the right-hand side of formula (48) are plotted along the axis of ordinates.

The values of each of the terms of formula (48) for three positions of the plunger are calculated in Table 7 to provide data for plotting a curve showing the variation in pressure under the plunger during the suction stroke. The curves plotted on the basis of these data are shown in Fig. 33*a*.

Curve *IV*, representing the loss of head h_s in the foot valve, is based on the fact that the highest value h_{s0} is reached when the valve lifts from its seat and then remains almost constant during the rest of the plunger stroke.

Adding the ordinates of the various curves thus plotted we can obtain the points of the summary curve *VII* which represents the variation of the value $\frac{p_s}{\gamma}$ with the plunger travel x and this curve can be obtained by drawing a line through these points. The vertical intercepts between the curves $\frac{p_s}{\gamma}$ and $\frac{p_a}{\gamma}$ represent the loss in head in the suction line, for the various positions of the pump plunger, in the scale adopted for the curves. The chart shown in Fig. 33*a* thus enables the effect of each term of formula (48) on the final value of the head $\frac{p_s}{\gamma}$ to be estimated.

In normal pump operation the liquid moves directly after the plunger and the value $\frac{p_s}{\gamma}$ must be greater than $\frac{p_v}{\gamma}$, the head of the saturated vapour of the liquid being pumped at the given temperature.

Table 7

Plunger position and velocity	Length of plunger travel x_s	Values of the terms in formula (48) for various positions of the plunger							
		$\frac{p_a}{\gamma}$	$z_s + x_s$	$(w_s+1)\frac{r^2\omega^2}{2g}\left(\frac{2x_s}{r}-\frac{x_s^2}{r^2}\right)$	h_s	$\frac{L_s}{g}r\omega^2\left(1-\frac{x_s}{r}\right)$	$\frac{x_s}{g}r\omega^2\left(1-\frac{x_s}{r}\right)$	$\frac{p_s}{\gamma}$
		I	II	III	IV	V	VI	VII
Initial position. Velocity $c=0$	$x_s = 0$	$\frac{p_a}{\gamma}$	$z_s + 0$	0	h_{s0}	$\frac{L_s}{g}r\omega^2$	0	$\frac{p_a}{\gamma}-\left(z_s + h_{s0} + \frac{L_s}{g}r\omega^2\right)$
Central position. Velocity $c=$ maximum	$x_s=r$	$\frac{p_a}{\gamma}$	$z_s + r$	$(w_s + 1)\frac{r^2\omega^2}{2g}$	h_s	0	0	$\frac{p_a}{\gamma}-\left[(z_s + r) + h_s + (w_s + 1)\frac{r^2\omega^2}{2g}\right]$
Final position. Velocity $c=0$	$x_s=2r$	$\frac{p_a}{\gamma}$	$z_s + 2r$	0	h_s	$-\frac{L_s}{g}r\omega^2$	$-\frac{2r^2\omega^2}{g}$	$\frac{p_a}{\gamma}-\left[(z_s + 2r)+h_s - \frac{L_s}{g}r\omega^2 - 2\frac{r^2\omega^2}{g}\right]$
Signification of the terms of formula (48)		Head on the surface of liquid level	Geodetic, or elevation, suction lift	Loss in head due to hydraulic resistances of the suction line	Loss of head in the foot valve	Inertia head in the suction line	Inertia head in the pump cylinder	Pressure head acting on the end of the plunger

If the value of $\frac{p_s}{\gamma}$, calculated by formula (48), is less than $\frac{p_v}{\gamma}$, the flow of liquid will separate from the bottom of the plunger. This reduces the pump capacity and causes a water hammer when the liquid runs up against the plunger again. If this occurs during the suction stroke and the resulting head in the cylinder exceeds that in the suction line, liquid will be forced into the discharge

Fig. 33.

line during this stroke. At best, this phenomenon will cause negative slip (overdelivery) of the pump, thereby impairing normal delivery of the liquid to its destination. At the worst, parts of the pumping installation may be broken. Water hammer occurring under these conditions is particularly dangerous when the velocities of the liquid and the plunger are directly opposite to each other.

The temperature of the liquid being handled can be taken into consideration by including the vapour pressure head $\frac{p_v}{\gamma}$ in the formula. Then

$$\frac{p_s}{\gamma} > \frac{p_a}{\gamma} - \left[\frac{p_v}{\gamma} + (z_s + x_s) + (1 + w_s)\frac{r^2\omega^2}{2g}\left(\frac{2x_s}{r} - \frac{x_s^2}{r^2}\right) + h_s + \frac{L_s}{g}r\omega^2\left(1 - \frac{x_s}{r}\right) + \frac{x_s}{g}r\omega^2\left(1 - \frac{x_s}{r}\right)\right] \quad (49)$$

It is evident from equation (49) that the higher the temperature of liquid being drawn into the pump, the poorer the suction conditions will be. Therefore, a pump handling heated liquids should be installed so that it precedes the heater while one for cooled liquids should follow the cooler if the vapour pressure is sufficiently low after cooling. Condensate and wet-air pumps of the condensing plant should be located so that they are supplied with liquid which flows by gravity (negative suction lift). If the absolute suction lift obtained by calculation is negative, the pumping installation has not been properly designed and various elements of the design must be altered until an acceptable value is obtained for the suction lift, but not less than 0.5 or 1 m of water column.

It is clear from the above that the operation of a pumping installation depends not only upon the pump but upon the piping as well.

It should be noted that in more precise calculations the actual length of the connecting rod must be taken into consideration. The piston velocity of direct-acting pumps can be assumed to be constant during the whole length of the stroke. Therefore, the inertia forces can be neglected in practical calculations.

Equation (49) can be applied to determine the maximum shaft speed of a power pump at which normal suction is ensured.

Since the value $\frac{p_s}{\gamma}$ must be greater than zero to provide normal suction, on the basis of equation (49), we can write

$$0 \leqslant \frac{p_a}{\gamma} - \left[\frac{p_v}{\gamma} + (z_s + x_s) + (1 + w_s)\frac{r^2\omega^2}{2g}\left(\frac{2x_s}{r} - \frac{r^2}{x_s^2}\right) + \right.$$
$$\left. + h_s + \frac{L_s + x_s}{g} r\omega^2 \left(1 - \frac{x_s}{r}\right)\right]$$

The minimum suction lift will be at the beginning of the plunger stroke when $x_s = 0$ or

$$0 \leqslant \frac{p_a}{\gamma} - \left[\frac{p_v}{\gamma} + \left(z_s + h_{so} + \frac{L_s}{g} r\omega^2\right)\right]$$

However, since $\omega = \frac{2\pi n}{60}$

then

$$0 \leqslant \frac{p_a}{\gamma} - \left[\frac{p_v}{\gamma} + z_s + h_{so} + \frac{L_s}{g}\left(\frac{2\pi n}{60}\right)^2 r\right] \qquad (50)$$

and

$$n \leqslant \frac{60\sqrt{g}}{2\pi}\sqrt{\frac{\frac{p_a}{\gamma} - \left[(z_s + h_{so}) + \frac{p_v}{\gamma}\right]}{L_s r}}$$

If we assume that $\frac{\sqrt{g}}{\pi} \approx 1$, we obtain

$$n \leqslant 30\sqrt{\left[\frac{p_a}{\gamma} - \left(\frac{p_v}{\gamma} + z_s + h_{so}\right)\right]\frac{1}{L_s r}} \qquad (51)$$

The maximum height $z_{s\,max}$ at which the pump can be installed above the surface of the liquid being pumped may be determined from equation (50). If it is assumed that $z_s = z_{s\,max}$, then

$$z_{s\,max} \leqslant \frac{p_a}{\gamma} - \left[\frac{p_v}{\gamma} + h_{so} + \frac{L_s}{g}\left(\frac{\pi n}{30}\right)^2 r\right]$$

The conclusions reached in the preceding discussion, enabling us to determine the variation in the suction lift during the suction stroke, refer to pumps in which the velocity of liquid flow in the piping varies from zero to its maximum value.

Such pumps include the single- and double-acting types.

The velocity of liquid flow in the suction lines of triplex single-acting, duplex double-acting and other multiple-cylinder pumps is much more uniform and does not reach zero values. This is due to the superimposed discharge of various ends of the cylinders.

It follows from this that the inertia forces are reduced when multiple-cylinder pumps are employed and, consequently, the variation in the head of liquid acting on the piston end during the suction stroke is likewise reduced.

(B) DISCHARGE STROKE OF THE PISTON

The energy required to deliver the liquid to its destination on the discharge line is imparted by the direct pressure of the piston (or plunger) on the liquid drawn into the cylinder during the suction stroke.

The change in the energy of the flow of liquid along its length of travel l_d in the discharge line when the flow velocity is v_d can be expressed by the equation

$$\int_{l_d}\left(z + \frac{p}{\gamma} + \frac{v_d^2}{2g}\right)dl_d + \frac{1}{g}\int_{l_d}\frac{\partial v_d}{\partial t}\,dl_d + $$
$$+ \sum_{l_d}\frac{\partial \zeta_d}{\partial l_d}dl_d = 0. \qquad (52)$$

Let us use the following notation for the discharge line:

$x_d x_d$ = plane of the extreme upper position of the plunger

p_d = variable pressure exerted by the liquid on the plunger during the discharge stroke, kg per sq m

p_o=pressure of the liquid at the outlet (point of delivery), kg per sq m
z_d=static head (geodetic height of discharge), m
f_{id}=area of the clear opening in the discharge pipe at various points, sq m
d_{id}=inside diameter of the discharge pipe at various points, m
v_o=velocity of liquid flow from the outlet orifice of the discharge line, m per sec
λ_{id}=coefficient of friction losses upon movement of the liquid in the pipeline
ζ_{id}=coefficient of local resistance.

Subtracting from the stroke S the length of travel x_d of the plunger from its upper position (Fig. 32), we obtain the remaining length of plunger travel $S-x_d$ to the end of the stroke. Then the first term of equation (52) can be written as follows:

$$\int_{l_d}\frac{\partial}{\partial l_d}\left(z+\frac{p}{\gamma}+\frac{v_d^2}{2g}\right)dl_d=\int_{S-x_d}^{z_d}\frac{\partial}{\partial l_d}(z)\,dl_d+$$

$$+\int_{p_d}^{p_o}\frac{\partial}{\partial l_d}\left(\frac{p}{\gamma}\right)dl_d+\int_{c}^{v_o}\frac{\partial}{\partial l_d}\left(\frac{v_d}{2g}\right)^2 dl_d=$$

$$=\int_{S-x_d}^{z_d}dz+\frac{1}{\gamma}\int_{p_d}^{p_o}dp+\frac{1}{2g}\int_{c}^{v_o}dv^2=$$

$$=z_d-(S-x_d)+\frac{p_o}{\gamma}-\frac{p_d}{\gamma}+\frac{v_o^2}{2g}-\frac{c^2}{2g}$$

Taking into consideration that the initial length of travel $S-x_d$ of the pumped liquid is along the cylinder, and solving the integral expressing the variation in the inertia forces by the same method used to determine the pressure on the plunger during its suction stroke, we obtain

$$\frac{1}{g}\int_{l_d}\frac{\partial v_d}{\partial t}dl_d=\sum_{l_d}\frac{1}{g}\int_{l_d}\frac{\partial v_{id}}{\partial t}dl_{id}=$$

$$=\frac{1}{g}\sum_{l_d}\int_{l_{id}}\frac{F}{f_{id}}\frac{\partial c}{\partial t}dl_{id}=\frac{1}{g}\left(\int_{S-x_d}\frac{F}{F}\frac{\partial c}{\partial t}dl_{S-x_d}+\right.$$

$$\left.+\int_{l_{1d}}\frac{F}{f_{1d}}\frac{\partial c}{\partial t}dl_{1d}+\int_{l_{2d}}\frac{F}{f_{2d}}\frac{\partial c}{\partial t}dl_{2d}+\dots\right)=$$

$$=\frac{1}{g}\left[(S-x_d)+\frac{F}{f_{1d}}l_{1d}+\frac{F}{f_{2d}}l_{2d}+\dots\right]\frac{\partial c}{\partial t}=$$

$$=\frac{1}{g}\left[(S-x_d)+\frac{D^2}{d_{1d}^2}l_{1d}+\frac{D^2}{d_{2d}^2}l_{2d}+\dots\right]\frac{\partial c}{\partial t}=$$

$$=\frac{L_d+(S-x_d)}{g}\frac{\partial c}{\partial t}$$

where $f_{1d}, f_{2d}, \dots$=area of the clear opening in the discharge pipeline over the lengths $l_{1d}, l_{2d}, \dots$, and having the diameters $d_{1d}, d_{2d}, \dots$.

$\left(\frac{F}{f_{1d}}l_{1d}+\frac{F}{f_{2d}}l_{2d}+\dots\right)=\left(\frac{D^2}{d_{1d}^2}l_{1d}+\frac{D^2}{d_{2d}^2}l_{2d}+\dots\right)=L_d$=referred length of the discharge pipeline. The loss of head in the discharge line, taking into account the hydraulic resistance h_d of the discharge valve at $v_{id}=\frac{F}{f_{id}}c$, is

$$\sum_{l_d}\frac{\partial\zeta_d}{\partial l_d}dl_d=\sum_{l_d}\lambda_{id}\frac{l_{id}}{d_{id}}\frac{v_{id}^2}{2g}+\sum_{l_d}\zeta_{id}\frac{v_{id}^2}{2g}+h_d=$$

$$=\frac{c^2}{2g}\left[\sum_{l_d}\left(\frac{F}{f_{id}}\right)^2\lambda_{id}\frac{l_{id}}{d_{id}}+\sum_{l_d}\left(\frac{F}{f_{id}}\right)^2\zeta_{id}\right]+h_d=$$

$$=\frac{c^2}{2g}\left[\sum_{l_d}\left(\frac{D}{d_{id}}\right)^4\lambda_{id}\frac{l_{id}}{d_{id}}+\sum_{l_d}\left(\frac{D}{d_{id}}\right)^4\zeta_{id}\right]+h_d=$$

$$=w_d\frac{c^2}{2g}+h_d$$

Here the expression within the brackets is denoted as w_d and is called the sum of the referred coefficients of resistance of the discharge line.

Substituting the results of integration into equation (52) we obtain

$$z_d-(S-x_d)+\frac{p_o}{\gamma}-\frac{p_d}{\gamma}+\frac{v_o^2}{2g}-\frac{c^2}{2g}+$$

$$+\frac{L_d+S-x_d}{g}\frac{\partial c}{\partial t}+w_d\frac{c^2}{2g}+h_d=0$$

Consequently, the pressure head of liquid acting on the plunger is equal to

$$\frac{p_d}{\gamma}=\frac{p_o}{\gamma}+(z_d-S+x_d)+\frac{c^2}{2g}(w_d-1)+$$

$$+\frac{v_o^2}{2g}+h_d+\frac{L_d+S-x_d}{g}\frac{\partial c}{\partial t}\qquad(53)$$

The rate of flow from an outlet orifice with an area f_o is equal to $v_o = \frac{F}{f_o} c$, therefore

$$\frac{p_d}{\gamma} = \frac{p_o}{\gamma} + (z_d - S + x_d) + \frac{c^2}{2g}(w_d - 1) + \frac{F^2}{f_o^2}\frac{c^2}{2g} + h_d + \frac{L_d + S - x_d}{g}\frac{\partial c}{\partial t} = \frac{p_o}{\gamma} + (z_d - S + x_d) + \frac{c^2}{2g}\left(w_d - 1 + \frac{F^2}{f_o^2}\right) + h_d + \frac{L_d + S - x_d}{g}\frac{\partial c}{\partial t} \quad (54)$$

Expressing the velocity of the plunger as a function of its length of travel and the angular velocity of crankpin rotation, we can write

$$\frac{p_d}{\gamma} = \frac{p_o}{\gamma} + \left[(z_d - S + x_d) + \left(w_d - 1 + \frac{F^2}{f_o^2}\right)\frac{r^2\omega^2}{2g}\left(\frac{2x_d}{r} - \frac{x_d^2}{r^2}\right) + h_d + \frac{L_d}{g} r\omega^2\left(1 - \frac{x_d}{r}\right) + \frac{S - x_d}{g} r\omega^2\left(1 - \frac{x_d}{r}\right)\right] \quad (55)$$

The values of each term in equation (55) are calculated in Table 8 for various positions of the plunger during the discharge stroke and are plotted in the chart (Fig. 33*b*). The curves representing the terms enable the effect of each term on the dynamic (discharge) head to be estimated. The travel of the plunger is plotted along the axis of abscissas and the corresponding values of the terms in equation (55) are plotted along the axis of ordinates.

The curve representing the loss of head in the discharge valve is based on the fact that at the beginning of the discharge stroke the head lost in lifting the valve from its seat is expended in overcoming the inertia of the valve itself, the inertia of a column of liquid above the valve in the discharge line and the head in the vessel into which the liquid is delivered.

This loss in head drops considerably after the valve has been lifted from its seat, and remains almost constant during the rest of the stroke.

Adding the ordinates of the various curves thus plotted we obtain the points of a summary curve which represents the variation in the value $\frac{p_d}{\gamma}$ (dynamic head) with the position of the plunger in the discharge stroke.

The form of the curve $\frac{p_d}{\gamma}$ shows that the head drops sharply at the end of the discharge stroke because of the effect of the inertia heads.

Table 8

Plunger position	Length of plunger travel, x_d	Values of the terms in formula (55)												Signification of the terms of formula (55)
		I	II	III	IV	V	VI	VII						
		$\frac{p_o}{\gamma}$	$z_d - (S - x_d)$	$\frac{r^2\omega^2}{2g}\left(w_d - 1 + \frac{F^2}{f_0^2}\right) \times \left(\frac{2x_d}{r} - \frac{x_d^2}{r}\right)$	h_d	$\frac{L_d}{g} r\omega^2\left(1 - \frac{x_d}{r}\right)$	$\frac{S - x_d}{g} r\omega^2\left(1 - \frac{x_d}{r}\right)$	$\frac{p_d}{\gamma}$						
Initial position	$x_d = 0$	$\frac{p_o}{\gamma}$	$z_d - S$	0	h_{do}	$\frac{L_d}{g} r\omega^2$	$\frac{S}{g} r\omega^2$	$\frac{p_{do}}{\gamma} + (z_d - S) + h_{do} + \frac{L_d}{g} r\omega^2 + \frac{S}{g} r\omega^2$						Pressure head acting on the end of the plunger
Central position	$x_d = r$	$\frac{p_o}{\gamma}$	$z_d - (S - r)$	$\frac{r^2\omega^2}{2g}\left(w_d - 1 + \frac{F^2}{f_0^2}\right)$	h_d	0	0	$\frac{p_o}{\gamma} + z_d - (S - r) + h_d + \frac{r^2\omega^2}{2g}\left(w_d - 1 + \frac{F^2}{f_0^2}\right)$						Inertia head in the pump cylinder
Final position	$x_d = 2r$	$\frac{p_o}{\gamma}$	z_d	0	h_d	$-\frac{L_d}{g} r\omega^2$	0	$\frac{p_o}{\gamma} + z_d + h_d - \frac{L_d}{g} r\omega^2$						Inertia head in the discharge line
		Head at the outlet orifice		Loss in head due to hydraulic resistances of the discharge line	Loss of head in the discharge valve									Geodetic, or elevation, head

A decrease in the head acting on the plunger during the discharge stroke may lead to the separation of the column of liquid from the bottom of the plunger, resulting in a pressure drop in the cylinder. Under such conditions the inlet valve may open and admit an additional volume of liquid from the suction line.

The water hammer that occurs when the plunger meets this additional volume of liquid may lead to a breakdown in the pump installation. Another unfavourable effect of this additional volume in the cylinder will be negative slip (overdelivery) of the pump.

An excessively low head may occur not only under the plunger but in various sections of the discharge line as well. This leads to a break in the flow of liquid and is called separation. The impact between the two parts of the column when they reunite may result in a water hammer that can cause damage to the line.

Therefore, whenever alterations are made in the piping of the shipboard pumping installation it is necessary to carry out suitable calculations to check the conditions of suction and discharge.

The preceding discussion concerns pumps in which the velocity of the liquid in the outlet piping drops periodically to zero during the discharge stroke. It follows from the capacity vs velocity curves that such conditions are inherent in simplex single- and double-acting types (Figs. 6 and 10).

In pumps with a higher discharge stroke coefficient (Figs. 12 and 13) the velocity of the liquid being pumped in the discharge line does not drop to zero and, therefore, the discharge pressure values will differ from those determined in the discussion above.

In such cases the discharge pressure of the pump is determined by plotting discharge curves for each of the cylinders and then adding the ordinates of the curves.

A maximum pump shaft speed, above which separation of the liquid flow may occur in the pipeline, exists for the discharge period as well as for the suction period.

Separation is most likely to occur at the end of the discharge stroke, at $x_d=2r$, when the pressure head under the plunger falls because the inertia heads, striving to break the column of liquid away from the plunger, reach their maximum values. Since the pressure head under the plunger must be positive to ensure normal discharge and since $\omega=\frac{2\pi n}{60}$, we can write on the basis of equation (55),

$$0 \leqslant \frac{p_o}{\gamma}+(z_d-S+x_d)+\left(w_d-1+\frac{F^2}{f_o^2}\right)\frac{r^2\omega^2}{2g}\left(\frac{2x_d}{r}-\frac{x_d^2}{r^2}\right)+$$
$$+h_d+\frac{L_d+S-x_d}{g}r\omega^2\left(1-\frac{x_d}{r}\right)$$

$$0 \leqslant \frac{p_o}{\gamma}+z_d+h_d+\frac{L_d}{g}r\omega^2=\frac{p_o}{\gamma}+z_d+h_d+\frac{L_d}{g}\left(\frac{2\pi n}{60}\right)^2$$

Assuming that $\frac{\sqrt{g}}{\pi}\approx 1$, then

$$n \leqslant \frac{60\sqrt{g}}{2\pi}\sqrt{\left(\frac{p_o}{\gamma}+z_d+h_d\right)\frac{1}{L_d r}}\approx$$
$$\approx 30\sqrt{\left(\frac{p_o}{\gamma}+z_d+h_d\right)\frac{1}{L_d r}} \qquad (56)$$

1-8. Vacuum and Air Chambers

(A) VACUUM CHAMBER IN THE SUCTION LINE

The suction line of a pump may be equipped with a vacuum chamber *1* to increase the pressure head on the bottom of the piston during the suction stroke by reducing the losses in head and the inertia heads in the suction line. This chamber (Fig. 34) divides the suction line into two parts. The first, long part *2* extends from the free surface of the liquid being pumped to the vacuum chamber while the second, short part *3* extends from the chamber to the pump.

Liquid enters the pump from the vacuum chamber if one is installed. The liquid level varies only slightly in the vacuum chamber during operation because of its considerable volume in comparison with the piston displacement. The large size of the vacuum chamber ensures a practically steady motion in the pipeline over the section from the free surface of the liquid to the chamber. This eliminates inertia heads in this part of the suction line.

Let us follow the action of a vacuum chamber as reflected by the capacity curve of a single-acting simplex pump on which it is installed (Fig. 35). The volume of liquid flowing from the vacuum chamber into the pump cylinder during a suction stroke of the piston should be equal to the volume entering the chamber during the suction and discharge strokes. Since the liquid flow up to the vacuum chamber can be considered to be practically steady, its rate of flow is constant. Accordingly, the area *ADBECA* is equal to the area *AHKGA*; the ordinates of the first area are equal to the liquid flow velocities in the short section of the suction line while those of the second area are equal to the steady flow velocity in the long section. At the beginning of the suction stroke the piston velocity equals zero and corresponds to point *A* of the chart. The mean velocity of liquid flow up to the chamber is proportional to ordinate *AG*. In the range of crankpin rotation from 0° to β_1° the steady velocity of

the liquid up to the chamber (ordinate AG) is greater than the piston velocity (ordinates between the lines AD and AN). Upon crankpin rotation through the angle β_1° the volume of liquid entering the vacuum chamber (area $AGDNA$) is greater than that drawn into the cylinder (area $ADNA$). Consequently, the level of the liquid in the chamber during this period rises and the air pressure in the chamber

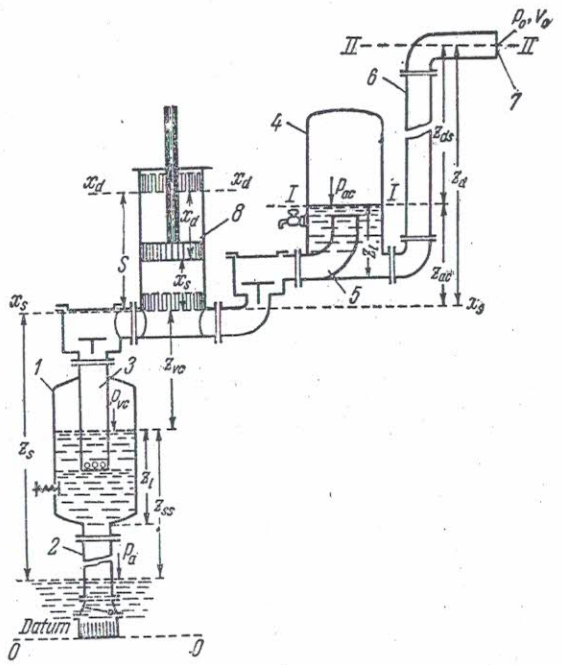


Fig. 34.

increases likewise. Upon further rotation of the crankpin from angle β_1° to β_2° the liquid level in the chamber drops as does the air pressure due to the increasing velocity of the piston (represented by the ordinates of the harmonic curve DBE).

In the range of crankpin rotation from angle β_1° to angle β_2° the volume of liquid entering the chamber corresponds to the area $NDELN$ and the volume drawn from the chamber into the cylinder—to the area $NDBELN$. When the crankpin reaches the angle β_2° the velocities of the liquid in the suction line and the piston become equal (point E in the chart). The piston velocity decreases as the crankpin continues to rotate from angle β_2° and this decreases the volume of

liquid drawn from the chamber into the cylinder. The rate at which the liquid enters the vacuum chamber remains constant. Thus, the liquid level in the chamber rises again. The suction stroke ends when the crankpin reaches the angle β_3° and the discharge stroke begins. During the discharge stroke of the piston the liquid level in the chamber rises, compressing the air whose pressure increases. This will continue all through the discharge stroke and until the crankpin has turned through the angle β°_1 in the suction stroke.

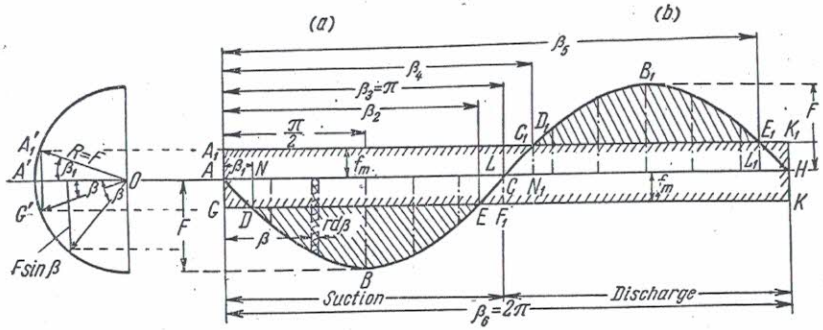


Fig. 35.

The volume of liquid entering the vacuum chamber during the discharge stroke corresponds to the area CF_1KHC.

Let us determine the pressure head under the piston during the suction stroke for a pump equipped with a vacuum chamber in the suction line. The following notation will be employed:

p_{vc} = air pressure in the vacuum chamber, kg per sq m
z_{ss} = distance from the free surface of the liquid being pumped to its mean level in the vacuum chamber, m
v_{ss} = velocity of steady liquid flow in the suction line up to the vacuum chamber, m per sec
l_{ss} = length of the suction line from the free surface of the liquid to the chamber, m
ζ_{ss} = sum of the referred coefficients of resistance of the section of the suction line from the free surface of the liquid to the chamber
L_{vc} = referred length of the piping from the vacuum chamber to the pump, m
w_{vc} = sum of the referred coefficients of resistance of the section of the suction line from the chamber to the pump
z_{vc} = distance from the mean level of liquid in the vacuum chamber to the lower end of the piston in its extreme lower position, m
h_s = loss of head in the inlet (foot) valve, m.

Since the liquid enters the cylinder from the vacuum chamber, the pressure head on the piston during the suction stroke, according to formula (47), is equal to

$$\frac{p_s}{\gamma}=\frac{p_{vc}}{\gamma}-\left[(z_{vc}+x_s)+(w_{vc}+1)\frac{c^2}{2g}+\right.$$
$$\left.+h_s+\frac{L_{vc}+x_s}{g}\frac{\partial c}{\partial t}\right] \qquad (57)$$

Expressing $\frac{p_{vc}}{\gamma}$ as a function of the head $\frac{p_a}{\gamma}$ on the free surface of the liquid being pumped, we obtain

$$\frac{p_{vc}}{\gamma}=\frac{p_a}{\gamma}-\left(z_{ss}+\sum_{l_{ss}}\zeta_{ss}\frac{v_{ss}^2}{2g}\right) \qquad (58)$$

Substituting equation (58) into formula (57) we can write

$$\frac{p_s}{\gamma}=\frac{p_a}{\gamma}-\left(z_{ss}+\sum_{l_{ss}}\zeta_{ss}\frac{v_{ss}^2}{2g}\right)-\left[(z_{vc}+x_s)+\right.$$
$$\left.+(w_{vc}+1)\frac{c^2}{2g}+h_s+\frac{L_{vc}+x_s}{g}\frac{\partial c}{\partial t}\right]=$$
$$=\frac{p_a}{\gamma}-\left[(z_{vc}+z_{ss})+(w_{vc}+1)\frac{c^2}{2g}+h_s+\right.$$
$$\left.+\sum_{l_{ss}}\zeta_{ss}\frac{v_{ss}^2}{2g}+\frac{L_{vc}+x_s}{g}\frac{\partial c}{\partial t}\right]$$

Then

$$\frac{p_s}{\gamma}=\frac{p_a}{\gamma}-\left[(z_s+x_s)+(w_{vc}+1)\frac{c^2}{2g}+\right.$$
$$\left.+h_s+\sum_{l_{ss}}\zeta_{ss}\frac{v_{ss}^2}{2g}+\frac{L_{vc}+x_s}{g}\frac{\partial c}{\partial t}\right] \qquad (59)$$

The value of the sum of the terms within the brackets is decreased because of the short length of piping in which the liquid travels in accordance with the variation in piston velocity. Consequently, the pressure head under the piston is increased. This pressure head is also more uniform during the suction stroke due to the reduction in the inertia head.

(B) AIR CHAMBER IN THE DISCHARGE LINE

An air chamber *4* is installed in the discharge line (Fig. 34) to achieve more uniform discharge and to reduce the variations in pressure under the piston during the discharge stroke. Such a chamber is

arranged as near as possible to the pump cylinder and it divides the discharge line into two sections: short section *5* and long section *6*. The short section *5* is between the pump cylinder and the air chamber, while the long section *6* is between the chamber and outlet hole *7* to which the liquid is being pumped. Pump *8* discharges the liquid directly into air chamber *4* if one has been installed. The liquid is delivered from the chamber to its final destination by the pressure of the air compressed in the chamber. The large volume of the air chamber in comparison with the piston displacement ensures a practically steady motion of the liquid in the discharge line after the air chamber. The motion of the liquid in section *5* of the line will not be steady and its velocity will vary in accordance with the variation in piston velocity.

Let us consider the action of an air chamber using the pump capacity curve in Fig. 35*b*.

During crankpin rotation from angle β_3° to angle β_4° the liquid level in the air chamber falls since the volume of liquid delivered from the pump cylinder to the chamber is less than that delivered from the chamber to the point of destination. The air pressure in the air chamber falls with the liquid level. This is evident from the chart in which the volume discharged into the chamber during crankpin rotation from β_3° to β_4° and represented by the area CD_1N_1C is less than the volume of liquid delivered from the chamber into the discharge line, corresponding to area $C_1CN_1D_1C_1$. When the crankpin reaches angle β_4° the velocity of the liquid entering the air chamber, represented by the ordinate D_1N_1 of the harmonic curve, is equal to the mean velocity of the steady flow of liquid (at point D_1) which is represented by the ordinate HK_1 of the straight line A_1K_1. Therefore, at the moment when the crankpin has rotated through the angle β_4° the volumes of liquid discharged into the air chamber and from it are equal. Upon further rotation of the crankpin the velocity of the liquid entering the chamber will be greater than the mean velocity of the steady flow of liquid in the pipeline after the chamber and, therefore, the liquid level rises in the chamber. During crankpin rotation from angle β_4° to angle β_5° the volume of liquid discharged from the air chamber is represented by the area $N_1D_1E_1L_1N_1$ and the volume entering the chamber, by $N_1D_1B_1E_1L_1N_1$. Thus, the volume of liquid remaining inside the air chamber is equal to $D_1B_1E_1D_1$ (in the accepted scale). At point E_1, corresponding to crankpin rotation to angle β_5°, the velocity of the liquid entering the chamber is again equal to the velocity of steady flow in the discharge line beyond the chamber, and the air pressure in the chamber reaches its maximum value. From this moment on, the velocity of liquid entering the chamber will decrease while that of the liquid leaving the

chamber will remain constant. This leads to a fall in the liquid level in the chamber and a corresponding drop in the air pressure. The suction stroke begins when the crankpin reaches the angle β° and liquid discharge from the cylinder into the air chamber ceases. After this, liquid is delivered to its destination due to the pressure of the air on the surface of the liquid in the chamber. Thus, the installation of an air chamber in the discharge line considerably reduces nonuniformity of the pump delivery. In the following we shall determine the pressure head acting on the bottom of the piston in this case during the discharge stroke.

Since the pump discharges into the air chamber, according to equation (53) the pressure head under the piston during the discharge stroke can be expressed as

$$\frac{p_d}{\gamma}=\frac{p_{ac}}{\gamma}+\left[z_{ac}-(S-x_d)+(w_{ac}-1)\frac{c^2}{2g}+\right.$$
$$\left.+h_d+\frac{v_{ac}^2}{2g}+\frac{L_{ac}+(S-x_d)}{g}\frac{\partial c}{\partial t}\right] \qquad (60)$$

where p_{ac} = air pressure in the air chamber, kg per sq m
z_{ac} = vertical distance from the liquid level in the chamber to the lower position of the piston, m
v_{ac} = velocity of liquid flow in the air chamber, m per sec
L_{ac} = referred length of the section of pipeline from the pump cylinder axis to the outlet hole in the air chamber connection, m
w_{ac} = sum of the referred coefficients of resistance of the section of the pipeline from the pump axis to the outlet in the air chamber connection
v_{ds} = velocity of steady liquid flow in the discharge line, m per sec
ζ_{ds} = sum of the coefficients of resistance in the section of the discharge line from the liquid level in the air chamber to the outlet orifice of the line.
z_{ds} = height of the outlet orifice of the discharge line above the liquid level in the air chamber, m.

By means of Bernoulli's equation we can determine $\frac{p_{ac}}{\gamma}$, expressing it as a function of $\frac{p_o}{\gamma}$, the pressure head at the outlet orifice of the discharge line.

If the plane passing through cross section $x_s x_s$ (Fig. 34) is taken as the datum, Bernoulli's equation for the cross sections *I—I* and *II—II* is written as

$$z_{ac}+\frac{p_{ac}}{\gamma}+\frac{v_{ac}^2}{2g}=z_d+\frac{p_o}{\gamma}+\frac{v_o^2}{2g}+\sum_{l_{ds}}\zeta_{ds}\frac{v_{ds}^2}{2g}$$

Consequently,

$$\frac{p_{ac}}{\gamma}=\frac{p_o}{\gamma}+(z_d-z_{ac})+\frac{v_o^2}{2g}-\frac{v_{ac}^2}{2g}+\sum_{l_{ds}}\zeta_{ds}\frac{v_{ds}^2}{2g} \qquad (61)$$

Substituting equation (61) into equation (60) we obtain

$$\frac{p_d}{\gamma}=\frac{p_o}{\gamma}+(z_d-z_{ac})+\frac{v_o^2}{2g}-\frac{v_{ac}^2}{2g}+\sum_{l_{ds}}\zeta_{ds}\frac{v_{ds}^2}{2g}+$$
$$+(z_{ac}-S+x_d)+(w_{ac}-1)\frac{c^2}{2g}+h_d+\frac{v_{ac}^2}{2g}+$$
$$+\frac{L_{ac}+(S-x_d)}{g}\frac{\partial c}{\partial t}$$

Therefore

$$\frac{p_d}{\gamma}=\frac{p_o}{\gamma}+(z_d-S+x_d)+\left[(w_{ac}-1)\frac{c^2}{2g}+\right.$$
$$\left.+\sum_{l_{ds}}\zeta_{ds}\frac{v_{ds}^2}{2g}+h_d\right]+\frac{L_{ac}+(S-x_d)}{g}\frac{\partial c}{\partial t} \qquad (62)$$

If an air chamber is installed in the discharge line the pressure under the piston during the discharge stroke is reduced because the length of the section of the discharge line in which the liquid flow is not steady is considerably shortened. Consequently, the parts of the pump that are subjected to forces arising from this pressure can be made lighter in design and operate under better conditions. Another advantage is that the pressure under the piston is more uniform throughout the discharge stroke.

(C) PRESSURE HEAD CHART FOR THE SUCTION AND DISCHARGE STROKES OF A PUMP EQUIPPED WITH VACUUM AND AIR CHAMBERS

If the equations (59) and (62) are expressed in terms of the angular velocity of the pump shaft and the length of piston travel we obtain

$$\frac{p_s}{\gamma}=\underbrace{\frac{p_a}{\gamma}}_{\text{I}}-\left[\underbrace{(z_s+x_s)}_{\text{II}}+\underbrace{(w_{vc}+1)\frac{r^2\omega^2}{2g}\left(\frac{2x_s}{r}-\frac{x_s^2}{r^2}\right)+\sum_{l_{ss}}\zeta_{ss}\frac{v_{ss}^2}{2g}}_{\text{III}}+\underbrace{\frac{h_s}{\text{IV}}}+\right.$$
$$\left.+\underbrace{\frac{L_{vc}}{g}r\omega^2\left(1-\frac{x_s}{r}\right)}_{\text{V}}+\underbrace{\frac{x_s}{g}r\omega^2\left(1-\frac{x_s}{r}\right)}_{\text{VI}}\right] \qquad (63)$$

$$\frac{p_d}{\gamma}=\underbrace{\frac{p_o}{\gamma}}_{\text{I}}+\underbrace{(z_d-S+x_d)}_{\text{II}}+$$

$$+\underbrace{(w_{ac}-1)\frac{r^2\omega^2}{2g}\left(\frac{2x_d}{r}-\frac{x_d^2}{r^2}\right)+\sum_{l_{ds}}\zeta_{ds}\frac{v_{ds}^2}{2g}}_{\text{III}}+$$

$$+\underbrace{h_d}_{\text{IV}}+\underbrace{\frac{L_{ac}}{g}r\omega^2\left(1-\frac{x_d}{r}\right)}_{\text{V}}+$$

$$=\underbrace{\frac{S-x_d}{g}r\omega^2\left(1-\frac{x_d}{r}\right)}_{\text{VI}} \qquad (64)$$

The curves on the chart in Fig. 36*a* have been plotted from the values of the separate terms of equation (63) for the various positions of the piston. The Roman numerals identify the curves with the terms of the equation. A similar chart has been plotted in Fig. 36*b* for the terms of equation (64).

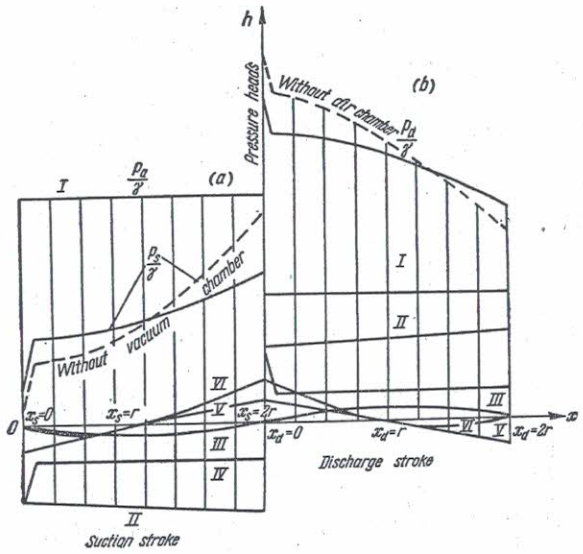


Fig. 36.

The dash lines in Fig. 36 show the variation in pressure head under the piston if the same simplex single-acting pump had no vacuum and air chambers. It is possible, by properly selecting the dimensions of the elements comprising pump and chamber design, to achieve an almost constant pressure under the piston throughout the stroke. In this case, lines $\frac{p_s}{\gamma}$ and $\frac{p_d}{\gamma}$ will be horizontal.

(D) VACUUM AND AIR CHAMBER DESIGN

The dimensions of a vacuum or air chamber depend upon the permissible nonuniformity of motion of the liquid in the suction line up to the vacuum chamber or in the discharge line after the air chamber.

The degree with which the liquid flow in the above-mentioned lines approaches a steady motion is evaluated by the corresponding nonuniformity of air pressure in the vacuum or air chamber. Thus

$$\delta_{vc}=\frac{p_{vc\,max}-p_{vc\,min}}{p_{vcm}} \text{ and } \delta_{ac}=\frac{p_{ac\,max}-p_{ac\,min}}{p_{acm}} \qquad (65)$$

where $p_{vc\,max}$, $p_{vc\,min}$ and $p_{vc\,m}$ = maximum, minimum and mean air pressure in the vacuum chamber during two strokes of the piston

$p_{ac\,max}$, $p_{ac\,min}$ and $p_{ac\,m}$ = maximum, minimum and mean air pressure in the air chamber during two strokes of the piston.

Let us determine the volume of a vacuum chamber installed in the suction line of a pump.

Denoting the maximum, minimum and mean volume of air in the chamber during two strokes of the piston as $V_{vc\,max}$, $V_{vc\,min}$ and $V_{vc\,m}$, we can write the following equation for the process of air compression and expansion according to Boyle's or Mariotte's law:

$$\frac{p_{vc\,max}}{p_{vc\,min}}=\frac{V_{vc\,max}}{V_{vc\,min}}$$

whence

$$\frac{p_{vc\,max}}{p_{vc\,min}}-1=\frac{V_{vc\,max}}{V_{vc\,min}}-1 \text{ or } \frac{p_{vc\,max}-p_{vc\,min}}{p_{vc\,min}}=\frac{V_{vc\,max}-V_{vc\,min}}{V_{vc\,min}} \qquad (66)$$

$$\frac{p_{vc\,max}}{p_{vc\,min}}+1=\frac{V_{vc\,max}}{V_{vc\,min}}+1 \text{ or } \frac{p_{vc\,max}+p_{vc\,min}}{p_{vc\,min}}=\frac{V_{vc\,max}+V_{vc\,min}}{V_{vc\,min}} \qquad (67)$$

Dividing equation (66) by equation (67)

$$\frac{p_{vc\,max}-p_{vc\,min}}{p_{vc\,max}+p_{vc\,min}}=\frac{V_{vc\,max}-V_{vc\,min}}{V_{vc\,max}+V_{vc\,min}} \qquad (68)$$

On the basis of equation (68) and since

$$\frac{p_{vc\,max}+p_{vc\,min}}{2}=p_{vc\,m} \text{ and } \frac{V_{vc\,max}+V_{vc\,min}}{2}=V_{vc\,m}$$

we obtain

$$\frac{p_{vc\,max}-p_{vc\,min}}{p_{vc\,m}}=\frac{V_{vc\,max}-V_{vc\,min}}{V_{vc\,m}}=\delta_{vc} \tag{69}$$

whence

$$V_{vc\,m}=\frac{V_{vc\,max}-V_{vc\,min}}{\delta_{vc}} \tag{70}$$

It must be noted that the same system of equations is valid for an air chamber which is installed in the discharge line.

We can now determine the required volume of a vacuum chamber installed in the suction line of a simplex single-acting pump. The variation in the volume of air in the chamber during two strokes of the piston (Fig. 35*a*) is:

$$\text{Area } DBED=\text{area } NDBELN-\text{area } NDELN=$$
$$=V_{vc\,max}-V_{vc\,min} \tag{71}$$

Since $f_m=\frac{F}{\pi}=0.318F$ (§ 1-4, p. 40) and since $f_m=F\sin\beta_1^\circ$ (Fig. 35) we can write

$$\sin\beta_1^\circ=\frac{f_m}{F}=\frac{0.318F}{F}=0.318 \text{ or } \beta_1^\circ=18.5^\circ \tag{72}$$

The length of the arc described by a crankpin with the radius r as it rotates through the angle β_1° will be:

$$r\beta_1^\circ=\frac{\pi r\times 18.5^\circ}{180^\circ}=0.103\pi r \tag{73}$$

According to Fig. 35*a*, the area $NDBELN=$

$$=2\int_{\beta_1^\circ}^{\frac{\pi}{2}} Fr\sin\beta d\beta \text{ and the area } NDELN=$$

$$=f_m r\,(\pi-2\beta_1^\circ) \text{ while } \beta_3^\circ-\beta_2^\circ=\beta_1^\circ$$

According to equations (71), (72) and (73)

$$V_{vc\,max}-V_{vc\,min}=2\int_{\beta_1^\circ}^{\frac{\pi}{2}} Fr\sin\beta d\beta-f_m r\,(\pi-2\beta_1^\circ)=$$

$$=2Fr\cos\beta_1^\circ-0.318F\,(\pi r-2\times 0.103\pi r)=1.1Fr=0.55FS \tag{74}$$

On the basis of equation (74) we find that for a single-acting simplex pump

$$V_{vc\,m}=\frac{V_{vc\,max}-V_{vc\,min}}{\delta_{vc}}=\frac{0.55FS}{\delta_{vc}} \text{ cu m} \tag{75}$$

The value $V_{vc\,max}-V_{vc\,min}$ and, consequently, $V_{vc\,m}$ can be found in a similar manner for pumps with other discharge stroke coefficients, thereby enabling us to design vacuum chambers for these pumps.

Since the theoretical volume of liquid entering the vacuum chamber (in the suction line) is equal to that entering the air chamber (in the discharge line), equations (70) and (75) can be used to determine $V_{ac\,m}$ in designing an air chamber.

The longer the pipe from the vacuum chamber to the cylinder and the higher the geodetic suction lift (static lift) the less the degree of pressure variation (nonuniformity), δ_{vc}, should be in the vacuum chamber. Data compiled by the Borets Pump Works (U.S.S.R.) on the permissible dynamic suction lift h_{dl} are given in Table 9.

Table 9

Pump shaft speed, n, rpm	Dynamic suction lift, $h_{dl}=z_s+h_{losses}$, m						
	Temperature of water being pumped, C°						
	0°	20°	30°	40°	50°	60°	70°
50	7.0	6.5	6.0	5.5	4.0	2.5	0
60	6.5	6.0	5.5	5.0	3.5	2.0	0
90	5.5	5.0	4.5	4.0	2.5	1.0	0
120	4.5	4.0	3.5	3.0	1.5	0.5	0
150	3.5	3.0	2.5	2.0	0.5	0	0
180	2.5	2.0	1.5	1.0	0	0	0

The value of δ_{vc} for vacuum chambers usually ranges from 0.02 to 0.05 while δ_{ac} for air chambers varies from 0.01 to 0.05. Liquid flow in a pipeline is considered to be steady at values of δ_{ch} as high as 0.025.

Table 10 lists the principal data required to calculate the mean volume of air in vacuum and air chambers.

In the table we have used the subscript *ch* to indicate that both vacuum and air chambers are implied.

Since the air occupies about 2/3 of the chamber volume, the total volume of the chamber should be

$$V_{ch}=\frac{3}{2}V_{ch\,m}=1.5V_{ch\,m} \text{ cu m} \tag{76}$$

If the chamber volume is too small, harmonic oscillations may develop in the liquid of the chamber due to the consecutive alterna-

Table 10

Type of pump	f_m	β_1°	$V_{ch\cdot max}-V_{ch\cdot min}$	$V_{ch\cdot m}=\frac{V_{ch\cdot max}-V_{ch\cdot min}}{\delta_{ch}}$		
				$\delta_{ch}=0.01$	$\delta_{ch}=0.02$	$\delta_{ch}=0.05$
Simplex single-acting	0.318	18.57	0.55 *FS*	55 *FS*	27.5 *FS*	11 *FS*
Simplex double-acting	0.637	39.53	0.21 *FS*	21 *FS*	10.5 *FS*	4.2 *FS*
Triplex single-acting	0.955	72.73	0.01 *FS*	1 *FS*	0.5 *FS*	0.2 *FS*
Duplex double-acting	1.273	19.00	0.04 *FS*	4 *FS*	2 *FS*	0.8 *FS*

tion of suction and discharge. These oscillations lead to resonance conditions if their frequency coincides with the number of pump shaft revolutions or piston strokes per minute. Resonance, in turn, greatly increases the amplitude of oscillation of the liquid level and the pressure in the chambers, thereby changing the steady flow in the pipelines, increasing the danger of water hammer and leading to other faults that impair normal pump operation.

Therefore, the chamber volume obtained using formula (76) should be checked for resonance conditions by the following formula:

$$V_{ch} > 2\left(\frac{3{,}000}{n_u}\right)^2 \frac{f_{chc}p_{ch\,max}}{l_{chc}+\frac{f_{chc}}{f_{ch}}z_{ch}} \text{ cu m} \qquad (77)$$

where n_u = number of pump strokes undulating the liquid per min
z_{ch} = height of the column of liquid in the chamber, m
l_{chc} = length of travel of the liquid from the pump cylinder axis to the hole in the pump connection, m
f_{chc} = cross-sectional area of the chamber connection, sq m
f_{ch} = cross-sectional area of the chamber, sq m
$p_{ch\,max}$ = maximum absolute pressure in the chamber, kg per sq m.

The number of pump strokes undulating the liquid per minute depends upon the discharge stroke coefficient k:

simplex single-acting ($k=1$) . . . $n_u=n$
simplex double-acting ($k=2$) . . . $n_u=2n$
triplex single-acting ($k=3$) . . . $n_u=3n$
duplex double-acting ($k=4$) . . . $n_u=4n$

where n is the pump shaft speed in rpm.

Vacuum and air chambers are available in a great variety of shapes but cylindrical chambers are most widely used in practice.

The diameter D_{ch} and height H_{ch} of a cylindrical chamber are:

$$D_{ch}=0.97\sqrt[3]{V_{ch}} \text{ m and } H_{ch}=1.36\sqrt[3]{V_{ch}} \text{ m}$$

Liquid delivery from the pump cylinder to the air chamber should be designed so that all the liquid passes through the chamber, changing its direction of flow and losing its velocity.

In the course of pump operation the amount of air, vapours and gases increases in the vacuum chamber since they are evolved from the liquid being pumped due to the low pressure in the chamber (partial vacuum). This accumulated air-vapour-gas mixture may be drawn in large portions into the pump cylinder and impair normal pump operation. This is avoided by providing small holes in the suction connection of the vacuum chamber, arranged below the liquid level (see Fig. 34). Such holes eliminate the air in small portions.

The volume of air in an air chamber decreases continuously in the course of pump operation due to the high pressure (it is absorbed by the liquid) and must be replenished through an air cock arranged below the suction valve of the pump. During the suction stroke this make-up air is drawn through the suction valve into the cylinder and during the discharge stroke is discharged into the air chamber. In high-pressure pumps make-up air is delivered to the air chamber at the required pressure through an air valve.

Excess pressure may develop in the air chamber when a pump is started. This is due to the necessity of imparting the initial acceleration (inertia head) to the liquid.

If the pressure in the air chamber of a pump standing idle is denoted as $p_{ac\,s}$ and that when the pump is started as $p_{ac\,max}$, then the coefficient of pressure increase is $\sigma=\frac{p_{ac\,max}}{p_{ac\,s}}$. Therefore,

$$p_{ac\,max}=\frac{p_{ac\,s}}{\sigma} \qquad (78)$$

The pressure $p_{ac\,max}$ is used in strength calculations of air chambers. According to equation (62)

$$p_{ac\,s}=p_o+z_d\gamma \qquad (79)$$

The coefficient of pressure increase σ can be found with the aid of the coefficient A which is calculated by the formula

$$A=\frac{l_d(FS)^2}{2gf_dV_{ac\,s}\frac{p_{ac\,s}}{\gamma}} \qquad (80)$$

where l_d = length of the discharge line beyond the air chamber, m
f_d = area of the clear opening in the discharge pipeline, sq m
v_{acs} = volume of air in the air chamber when the pump is standing idle, in cu m.

The values of σ as a function of A are given in Table 11.

Table 11

A	0.0000	0.0012	0.0044	0.0093	0.0156	0.0316	0.0408	0.0508	0.0512	0.0712	0.0834	0.1190
σ	1.0	1.05	1.1	1.15	1.2	1.3	1.35	1.4	1.45	1.50	1.55	1.70

The high inertia of the stationary liquid in the discharge line would make drive motor overloading inevitable in starting a pump if motion was imparted directly to the liquid and not through the elastic medium (air) in the air chamber.

1-9. Work Required to Drive the Pump Piston and Pump Efficiencies

Let us first determine the work done in its cycle of two strokes by the piston of a simplex single-acting power pump not equipped with vacuum and air cylinders. The results obtained can then be used to determine the work done in two strokes by the pistons of power pumps with any discharge stroke coefficient.

Since the nonoperative end of the cylinder in a single-acting pump is usually open to the atmosphere its pressure can be assumed to be atmospheric.

During the suction stroke the pressure head of the liquid acting on the piston is expressed by equation (48):

$$\frac{p_s}{\gamma}=\frac{p_a}{\gamma}-\left[(z_s+x_s)+(w_s+1)\frac{r^2\omega^2}{2g}\left(\frac{2x_s}{r}-\frac{x_s^2}{r^2}\right)+\right.$$
$$\left.+h_s+\frac{L_s}{g}r\omega^2\left(1-\frac{x_s}{r}\right)+\frac{x_s}{g}r\omega^2\left(1-\frac{x_s}{r}\right)\right]$$

The force R_s acting on the piston during the suction stroke will be

$$R_s=\gamma F\left(\frac{p_a}{\gamma}-\frac{p_s}{\gamma}\right)=\gamma F\left\{\frac{p_a}{\gamma}-\left[\frac{p_a}{\gamma}-(z_s+x_s)-\right.\right.$$
$$-(1+w_s)\frac{r^2\omega^2}{2g}\left(\frac{2x_s}{r}-\frac{x_s^2}{r^2}\right)-h_s-$$
$$\left.\left.-\frac{L_s}{g}r\omega^2\left(1-\frac{x_s}{r}\right)-\frac{x_s}{g}r\omega^2\left(1-\frac{x_s}{r}\right)\right]\right\} \qquad (81)$$

The work done by the forces applied to the piston during its travel over the length dx_s is

$$dA_s=\gamma F\left\{(z_s+x_s)+w_s\frac{r^2\omega^2}{2g}\left(\frac{2x_s}{r}-\frac{x_s^2}{r^2}\right)+h_s+\right.$$
$$\left.+\frac{L_s}{g}r\omega^2\left(1-\frac{x_s}{r}\right)+\left[\frac{x_s}{g}r\omega^2\left(1-\frac{x_s}{r}\right)+\frac{r^2\omega^2}{2g}\left(\frac{2x_s}{r}-\frac{x_s^2}{r^2}\right)\right]\right\}dx_s$$

The work done by the forces applied to the piston over the whole suction stroke is

$$A_s=\int_{x=0}^{x_s=2r}\gamma F\left\{(z_s+x_s)+w_s\frac{r^2\omega^2}{2g}\left(\frac{2x_s}{r}-\frac{x_s^2}{r^2}\right)+h_s+\right.$$
$$\left.+\frac{L_s}{g}r\omega^2\left(1-\frac{x_s}{r}\right)+\left[\frac{x_s}{g}r\omega^2\left(1-\frac{x_s}{r}\right)+\frac{r^2\omega^2}{2g}\left(\frac{2x_s}{r}-\frac{x_s^2}{r^2}\right)\right]\right\}dx_s \qquad (82)$$

We next divide equation (82) into the various components of work done by the piston in drawing the liquid into the cylinder, and find the values of the components.

The work done in lifting the liquid from its free surface to the upper position of the pump piston is

$$\gamma F\int_{x_s=0}^{x_s=2r}(z_s+x_s)\,dx_s=\gamma F\left[z_sx_s+\frac{x_s^2}{2}\right]_{x_s=0}^{x_s=2r}=$$
$$=\left(2rz_s+\frac{4r^2}{2}\right)\gamma F=(z_s+r)S\gamma F$$

Work done in overcoming hydraulic resistances in the suction line and the pump inlet valve is

$$\gamma F\int_{x_s=0}^{x_s=2r}\left[(w_s+1)\frac{r^2\omega^2}{2g}\left(\frac{2x_s}{r}-\frac{x_s^2}{r^2}\right)+h_s\right]dx_s=$$
$$=\left[(w_s+1)\frac{r^2\omega^2}{2g}\left(\frac{2x_s^2}{2r}-\frac{x_s^3}{3r^2}\right)+h_sx_s\right]_{x_s=0}^{x_s=2r}\gamma F=$$
$$=\left[\frac{4}{3}\frac{r^2\omega^2}{2g}(w_s+1)+2h_sr\right]\gamma F$$

The work per piston stroke of the inertia forces of the liquid flowing in the pipeline is

$$\gamma F\int_{x_s=0}^{x_s=2r}\frac{L_s}{g}r\omega^2\left(1-\frac{x_s}{r}\right)dx_s=\left[\frac{L_s}{g}r\omega^2\left(x_s-\frac{x_s^2}{2r}\right)\right]_{x_s=0}^{x_s=2r}\gamma F=0$$

The work per piston stroke of the inertia forces of the liquid filling the cylinder is

$$\gamma F \int_{x_s=0}^{x_s=2r} \frac{x_s}{g} r\omega^2 \left(1-\frac{x_s}{r}\right) dx_s = \left[\frac{r\omega^2}{g}\left(\frac{x_s^2}{2}-\frac{x_s^3}{3r}\right)\right]_{x_s=0}^{x_s=2r} \gamma F =$$
$$= \frac{r\omega^2}{g}\left(\frac{4r^2}{2}-\frac{8r^3}{3r}\right)\gamma F = -\frac{2}{3}\frac{r^3\omega^2}{g}\gamma F$$

This negative value of the work per piston stroke of the inertia forces of the liquid is explained by the fact that the mass of the liquid filling the pump cylinder during the first half of the stroke is less than at the end of the stroke. Therefore, the work done in overcoming the forces of inertia and in imparting acceleration to the liquid during the first half of the piston stroke is less than the work of the inertia forces facilitating the movement of the piston during the second half of its stroke.

On the basis of the preceding results of integration and equation (82) we can find the work done by the forces acting on the piston during its suction stroke:

$$A_s = \gamma F\left[(z_s+r)S+\frac{4}{3}\frac{r^3\omega^2}{2g}(w_s+1)+2h_s r-\frac{2}{3}\frac{r^3\omega^2}{g}\right] =$$
$$= \gamma F\left[(z_s+r)+\frac{4}{3}\frac{r^3\omega^2}{2g}w_s+2h_s r+\frac{4}{3}\frac{r^3\omega^2}{2g}-\frac{4}{3}\frac{r^3\omega^2}{2g}\right] =$$
$$= \gamma F\left[S\left(z_s+\frac{S}{2}\right)+\left(\frac{2}{3}\frac{r^2\omega^2}{2g}w_s+h_s\right)S\right]$$

Thus

$$A_s = \gamma F S\left[\left(z_s+\frac{S}{2}\right)+\frac{2}{3}\frac{r^2\omega^2}{2g}w_s+h_s\right] \text{ kg-m} \qquad (83)$$

The work done during the suction stroke by all the forces acting on the piston is graphically represented by the area A_s in Fig. 37*a*.

The work done during piston travel in the discharge stroke over the distance dx_d by the forces R_d acting on the piston is

$$dA_d = R_d\, dx_d = \gamma F\left(\frac{p_d}{\gamma}-\frac{p_a}{\gamma}\right)dx_d$$

Using equation (55) which determines the pressure head acting on the piston during the discharge stroke, we can write that the work done during the piston stroke (Fig. 32) is

$$A_d = \int_{x_d=0}^{x_d=2r} \gamma F\left\{\left[\frac{p_o}{\gamma}+(z_d-S+x_d)+\right.\right.$$

$$+\frac{r^2\omega^2}{2g}\left(\frac{2x_d}{r}-\frac{x_d^2}{r^2}\right)\left(w_d-1+\frac{F^2}{f_o^2}\right)+h_d+\frac{L_d}{g}r\omega^2\left(1-\frac{x_d}{r}\right)+$$
$$\left.\left.+\frac{S-x_d}{g}r\omega^2\left(1-\frac{x_d}{r}\right)\right]-\frac{p_a}{\gamma}\right\}dx_d \qquad (84)$$

As before, we divide equation (84) into its components which express the work done by the various forces acting on the piston and integrate them.

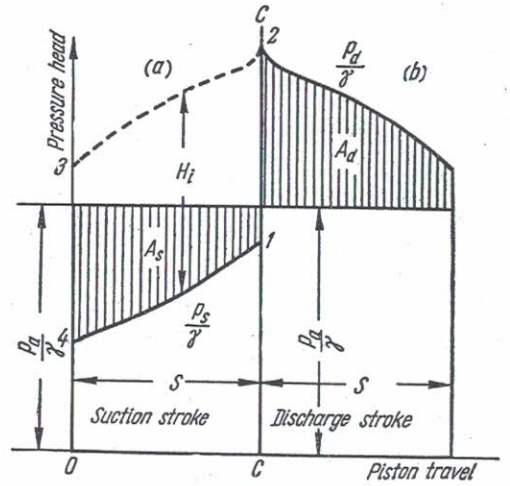


Fig. 37

The work done by the forces overcoming the pressure in the vessel into which the liquid is being pumped is

$$\gamma F \int_{x_d=0}^{x_d=2r}\left(\frac{p_o}{\gamma}-\frac{p_a}{\gamma}\right)dx_d = \left[\frac{p_o}{\gamma}-\frac{p_a}{\gamma}\right]_{x_d=0}^{x_d=2r} x_d \quad \gamma F = \left(\frac{p_o}{\gamma}-\frac{p_a}{\gamma}\right)S\gamma F$$

The work done in raising the liquid to the geodetic height of delivery (static head) is

$$\gamma F \int_{x_d=0}^{x_d=2r}(z_d-S+x_d)\,dx_d = \left[z_d x_d - S x_d + \frac{x_d^2}{2}\right]_{x_d=0}^{x_d=2r}\gamma F =$$
$$= \left(2rz_d-2rS+\frac{4r^2}{2}\right)\gamma F = \left(z_d-\frac{S}{2}\right)S\gamma F$$

The work done in overcoming hydraulic resistances in the discharge

line and in the discharge valve of the pump is

$$\gamma F \int_{x_d=0}^{x_d=2r} \left[\frac{r^2\omega^2}{2g}\left(\frac{2x_d}{r}-\frac{x_d^2}{r^2}\right)\left(w_d-1+\frac{F^2}{f_o^2}\right)+h_d\right]dx_d =$$

$$=\left(w_d-1+\frac{F^2}{f_o^2}\right)\frac{r^2\omega^2}{2g}\left[\left(\frac{2x_d^2}{2r}-\frac{x_d^3}{3r^2}\right)+h_d x_d\right]_{x_d=0}^{x_d=2r}\gamma F =$$

$$=\left[\frac{4}{3}\frac{r^2\omega^2}{2g}\left(w_d-1+\frac{F^2}{f_o^2}\right)+2rh_d\right]\gamma F =$$

$$=\left[\frac{2}{3}\frac{r^2\omega^2}{2g}\left(w_d-1+\frac{F^2}{f_o^2}\right)+h_d\right]\gamma FS$$

The work of the inertia forces of the liquid flowing in the pipeline during the discharge stroke is

$$\gamma F \int_{x_d=0}^{x_d=2r} \frac{L_d}{g} r\omega^2\left(1-\frac{x_d}{r}\right)dx_d = \gamma F \frac{L_d}{g} r\omega^2\left[x_d-\frac{x_d^2}{2r}\right]_{x_d=0}^{x_d=2r} = 0$$

The work of the inertia forces of the liquid filling the pump cylinder during the discharge stroke is

$$\gamma F \int_{x_d=0}^{x_d=2r} \frac{S-x_d}{g} r\omega^2\left(1-\frac{x_d}{r}\right)dx_d =$$

$$=\int_{x_d=0}^{x_d=2r}\frac{S}{g}r\omega^2\left(1-\frac{x_d}{r}\right)\gamma F\,dx_d-\int_{x_d=0}^{x_d=2r}\frac{x_d}{g}r\omega^2\left(1-\frac{x_d}{r}\right)\gamma F\,dx_d=$$

$$=\gamma F\left[\frac{S}{g}r\omega^2\left(x_d-\frac{x_d^2}{2r}\right)-\frac{r\omega^2}{g}\left(\frac{x_d^2}{2r}-\frac{x_d^3}{3r}\right)\right]_{x_d=0}^{x_d=2r}=$$

$$=\frac{1}{g}r\omega^2\left[\left(2r-\frac{4r^2}{2r}\right)S-\left(\frac{4r^2}{2r}-\frac{8r^3}{3r}\right)\right]\gamma F=\frac{2}{3}\frac{r^3\omega^2}{g}\gamma F$$

Adding the results of integration, we obtain according to equation (84):

$$A_d=\gamma F\left\{\left(\frac{p_o}{\gamma}-\frac{p_a}{\gamma}\right)S+\left(z_d-\frac{S}{2}\right)S+\right.$$

$$\left.+\left[\frac{2}{3}\frac{r^2\omega^2}{2g}\left(w_d-1+\frac{F^2}{f_o^2}\right)+h_d\right]S+\frac{2}{3}\frac{r^3\omega^2}{g}\right\}=$$

$$=\gamma F\left[S\left(\frac{p_o}{\gamma}-\frac{p_a}{\gamma}\right)+\left(z_d-\frac{S}{2}\right)S+\frac{2}{3}\frac{r^2\omega^2}{2g}\left(w_d+\frac{F^2}{f_o^2}\right)S-\right.$$

$$\left.-\frac{2}{3}\frac{r^2\omega^2}{2g}S+h_dS+\frac{2}{3}\frac{r^2\omega^2}{2g}S\right]$$

$$A_d=\gamma FS\left[\left(\frac{p_o}{\gamma}-\frac{p_a}{\gamma}\right)+\left(z_d-\frac{S}{2}\right)+\right.$$

$$\left.+\frac{2}{3}\frac{r^2\omega^2}{2g}\left(w_d+\frac{F^2}{f_o^2}\right)+h_d\right]\ \text{kg-m} \qquad (85)$$

The work done during the discharge stroke by all the forces acting on the piston is graphically represented by the area A_d (Fig. 37*b*).

The work done during the two strokes (suction and discharge) of the piston is

$$A_i=A_s+A_d=\gamma FS\left[\left(z_s+\frac{S}{2}\right)+\frac{2}{3}\frac{r^2\omega^2}{2g}w_s+h_s\right]+$$

$$+\gamma FS\left[\left(\frac{p_o}{\gamma}-\frac{p_a}{\gamma}\right)+\left(z_d-\frac{S}{2}\right)+\frac{2}{3}\frac{r^2\omega^2}{2g}\left(w_d+\frac{F^2}{f_o^2}\right)+h_d\right]$$

$$A_i=\gamma FS\left(\frac{p_o}{\gamma}-\frac{p_a}{\gamma}\right)+\left(z_s+z_d\right)+$$

$$+\frac{2}{3}\frac{r^2\omega^2}{2g}\left[\left(w_d+\frac{F^2}{f_o^2}+w_s\right)+(h_s+h_d)\right]\text{kg-m} \qquad (86)$$

Thus, the work done by the piston during two strokes is expended in:

(a) overcoming the pressure $\frac{p_o}{\gamma}-\frac{p_a}{\gamma}$ in the vessel to which the liquid is delivered by the pump;

(b) overcoming the hydraulic resistances in the suction and discharge lines $\frac{2}{3}\frac{r^2\omega^2}{2g}\left(w_d+\frac{F^2}{f_o^2}+w_s\right)=h_{dp}+h_{sp}$;

(c) overcoming the resistances of the inlet and discharge valves h_s+h_d.

The sum of the enumerated work constitutes the indicated work A_i of the piston.

The total indicated pressure head created by the piston is

$$H_i=\frac{p_i}{\gamma}=\left(\frac{p_o-p_a}{\gamma}+z_s+z_d\right)+(h_{sp}+h_{dp})+$$

$$+(h_s+h_d)+h_c=H_u+H_{lp}+H_c=H+H_c\ \text{mH}_2\text{O}, \qquad (87)$$

where $H_u=\frac{p_o}{\gamma}-\frac{p_a}{\gamma}+z_s+z_d$ = useful, or static, head lost in raising the liquid to the point of delivery, mH_2O

$H_{lp}=h_{sp}+h_{dp}$ = head lost in overcoming the resistances of the suction and discharge lines, mH_2O

$H_c=h_s+h_d+h_c$ = head lost in the valves (h_d+h_s) of the pump cylinder, and in the cylinder itself (h_c), mH_2O

$H = H_u + H_{lp}$ = head lost outside the cylinder and called the manometric head, mH_2O.

Combining equations (87) and (86) we obtain

$$A_i = \gamma FS(H_u + H_{lp} + H_c) = (H + H_c)\gamma FS \text{ kg-m} \quad (88)$$

If the area representing the work A_d in Fig. 37*b* is turned through 180° about the axis *cc* into the plane of area A_s (Fig. 37*a*), the theoretical indicator diagram 1, 2, 3 and 4 is obtained. The area of this indicator diagram corresponds to the indicated work A_i.

It is clear from equation (88) that the work done by the piston in two strokes is proportional to the displacement FS of the pump cylinder.

The volume of liquid delivered in two strokes of the piston, when the dimensions of the cylinder remain constant, is proportional to the discharge stroke coefficient. Thus, the work done by the piston of a pump with any discharge stroke coefficient k, in one revolution of the pump shaft, is

$$A_{ik} = k\gamma FSH_i \text{ kg-m} \quad (89)$$

The theoretical indicated power of the liquid cylinder of a pump is

$$N_{it} = \frac{A_{ik}n}{60} = \frac{k\gamma FSnH_i}{60} \text{ kg-m per sec} \quad (90)$$

Equation (90) does not include the various losses which characterize the quality of pump manufacture, and design and operational features of the pump. These qualities or features can be assessed by computing the pump efficiency. There are several kinds of pump efficiency. Let us determine the volumetric efficiency of a pump.

The amount of liquid that remains in the pump cylinder at the end of the suction stroke of the piston is less than the volume swept through or displaced by the piston in this stroke. The amount of liquid entering the discharge line during the discharge stroke is less than the volume of the liquid remaining in the cylinder after the suction stroke. Thus, due to these reasons, the actual discharge of a pump is always less than the theoretical displacement. During the suction stroke the liquid entering the cylinder does not fill it completely since the air, gases and vapours evolved from the liquid occupy a part of the cylinder volume. Clearances between the moving parts of the pump that separate the cylinder ends and the different pressures to which the liquid is subjected in the cylinder ends cause leakage through these clearances. Besides this, delayed closing and opening of the valves also leads to volumetric losses of the liquid. Indeed, when the piston reaches its upper dead point at the end of the suction stroke, the inlet valve does not close immediately since the column of liquid under its disk exerts a pressure equal to or higher than

that of the liquid in the cylinder. The valve begins to close only when the piston passes the dead point and starts its discharge stroke, thereby raising the pressure in the cylinder.

Before the inlet valve seats properly, the piston will have travelled a certain distance of its discharge stroke, forcing some of the liquid that entered the cylinder during the suction stroke back into the suction line. Therefore, by the time the discharge valve opens, a part of the volume of liquid meant for delivery is lost.

Liquid is drawn back into the cylinder from the discharge line, due to delayed closing of the discharge valve, in a similar manner. When the piston reaches the lower extreme position at the end of the discharge stroke, the discharge valve does not return immediately to its seat because it is held open by a certain volume of liquid under its disk and the inlet valve cannot open immediately when the suction stroke begins since the pressure in the cylinder is higher than that in the suction line. Only after the piston travels a certain distance of its suction stroke, creating a partial vacuum in the cylinder, will the discharge valve close and the suction valve open.

During this part of its stroke the piston will draw a certain amount of liquid from the discharge line through the still open discharge valve. Therefore, when the suction valve opens, a part of the cylinder volume will be occupied by liquid drawn back into the cylinder from the discharge line. This, in turn, affects the time lag with which the suction valve opens.

These volumetric losses, known as pump slip, are taken into consideration in the volumetric efficiency which was mentioned earlier in connection with the calculation of the actual capacity of a pump.

The volumetric efficiency

$$\eta_v = \frac{Q}{Q_t} = \frac{\text{actual volumetric capacity of the pump in unit time}}{\text{volume swept through by the piston in unit time}}. \quad (91)$$

The volumetric efficiency is the principal economical index characterizing the operation of a reciprocating pump.

Approximate values of the volumetric efficiency of well-made pumps handling water are listed in Table 12.

The curves in Fig. 38 indicate the variation of the volumetric efficiency with the kinematic viscosity of the pumped liquid at various pump shaft speeds. The efficiency values are given in per cent of the corresponding values for pumping water.

We shall call the indicated power of the liquid cylinder, calculated from the actual capacity of the pump, the actual indicated power of the pump:

$$N_i = \frac{\eta_v k\gamma FSnH_i}{60} \text{ kg-m per sec} \quad (92)$$

Table 12

Type and cylinder diameter of pump	Q, cu m per h	Volumetric efficiency η_v		Mean piston velocity c_m, m per sec
		New pumps	Used pumps	
Low-capacity power pumps—50 mm	Up to 20	0.85 to 0.95	0.8 to 0.85	0.2 to 0.5
Medium-capacity power pumps—50 to 150 mm	20 to 60	0.90 to 0.97	0.85 to 0.92	0.5 to 0.9
High-capacity power pumps—over 150 mm	60 and more	0.95 to 0.99	0.9 to 0.95	1 to 2
Direct-acting pumps	Any capacity	0.96 to 0.99	0.92 to 0.95	0.2 to 0.7
Hand pumps		0.90 to 0.95	0.85 to 0.9	0.05 to 0.15
Hand fire pumps		0.95 to 0.99	0.80 to 0.95	0.3 to 0.5

Notes: 1. The value of η_v depends upon the degree of wear of the pump parts and should be taken as $\eta_v = 0.4$ to 0.5 for worn-out pumps.
2. The values of η_v and c_m will be 5 to 10 per cent less than the tabular values if thick, viscous or heavily contaminated liquids are handled.

Then the volumetric efficiency will be

$$\frac{N_i}{N_{it}} = \frac{\frac{\eta_v k \gamma FSnH_i}{60}}{\frac{k\gamma FSnH_i}{60}} = \eta_v \approx 0.85 \text{ to } 0.99$$

It follows from this that the power expended in displacing the volumetric losses of liquid, taken into consideration in the volumetric efficiency, does no useful work.

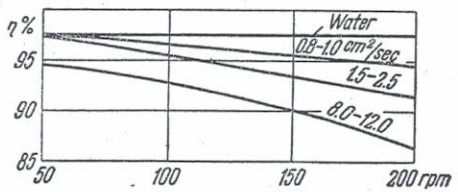


Fig. 38.

The power required to move the liquid from its lower level to the point of discharge, without regard for the volumetric losses and the losses of head in the liquid cylinder itself, shall be called the theoretical useful power N_{ut}. Then

$$N_{ut} = \frac{k\gamma FSHn}{60} \text{ kg-m per sec} \qquad (93)$$

Losses in head occur in the pump itself. They depend upon the design of the valves, finish of the machined surfaces over which the liquid flows and the magnitude of the local resistances along the flow. These losses are accounted for by the hydraulic efficiency η_h.

It follows that the hydraulic efficiency is the ratio of the total head (dynamic head plus dynamic lift) H, created by the pump, to the indicated head H_i, or the ratio of the power values producing these heads. Thus

$$\eta_h = \frac{H}{H_i} = \frac{H}{H + H_c} = \frac{k\gamma FSHn}{k\gamma FSH_i n} \approx 0.75 \text{ to } 0.98$$

If we take into account in the volumetric efficiency the reduction in the theoretical pump capacity due to various kinds of volumetric losses, we can obtain the equation for the actual useful power of the liquid cylinder:

$$N_u = \frac{\eta_v k\gamma FSnH}{60} \text{ kg-m per sec}$$

Then the indicated efficiency of the liquid cylinder will be

$$\eta_i = \frac{N_u}{N_{it}} = \frac{\frac{\eta_v k\gamma FSHn}{60}}{\frac{k\gamma FS(H + H_c)n}{60}} = \eta_v \eta_h \approx 0.64 \text{ to } 0.97 \qquad (94)$$

Taking into account the power losses due to friction in the parts of the liquid cylinder mechanism by introducing the mechanical efficiency η_m we can find the power developed by the pump motive unit on the pump shaft. This will be

$$N_m = \frac{N_{it}}{\eta_m}$$

whence

$$\eta_m = \frac{N_{it}}{N_m} \approx 0.85 \text{ to } 0.95 \qquad (95)$$

The overall efficiency of the liquid cylinder of a pump is the ratio of the actual useful power of the liquid cylinder to the power developed on the pump shaft:

$$\eta_{op} = \frac{N_u}{N_m} = \frac{N_u}{\frac{N_{it}}{\eta_m}} = \frac{N_u}{\frac{1}{\eta_m}\frac{N_u}{\eta_i}} = \eta_m \eta_i = \eta_v \eta_h \eta_m \approx 0.55 \text{ to } 0.92 \qquad (96)$$

The overall efficiency of the liquid cylinder enables the economy of pump operation to be assessed. The value of η_{op} depends both upon the quality of the pump and upon its size. It ranges from 0.55 to 0.92 (see Table 13).

Table 13

Type of pump	New power pumps		New direct-acting pumps
	Without gearing	With worm gear transmission	
Low-capacity (D≤50 mm)	0.65 to 0.75	0.55 to 0.65	0.8 to 0.83
Medium-capacity (D>50 to 150 mm)	0.75 to 0.85	0.65 to 0.75	0.83 to 0.85
High-capacity (D>150 mm)	0.85 to 0.92	0.75 to 0.8	0.85 to 0.92
Hand	0.5 to 0.85		

In small pumps, a considerable part of the work is expended in overcoming the friction in the stuffing boxes. Therefore, the overall efficiency of these pumps depends to a considerable degree upon the stuffing box design, the type of packing and its condition in operation.

The power required on the shaft of the pump motive unit is calculated by the formula

$$N_m = \frac{\eta_v k \gamma F S n H}{60 \times 75 \eta_{op}} = \frac{Q_s \gamma H}{75 \eta_{op}} \text{ metric hp} \qquad (97)$$

where Q_s = output of the pump, cu m per sec.

For various reasons the hydraulic resistances in the system and volumetric losses in the pump may increase when pumps are operated under shipboard conditions.

Moreover, leaks that are difficult to discover may appear in the suction line, as well as leakage through the piston packing, rod stuffing boxes, valves, gaskets, etc.

All this leads to a reduction in the delivery of liquid to the point of discharge. To ensure that the required volume of liquid is delivered it is necessary to provide for an increase in pump capacity by increasing the speed of the pump shaft.

For this purpose, reserve motive unit power is provided, depending upon the calculated power required on the pump shaft and of a value indicated in Table 14.

Table 14

Power required on pump shaft	Steam engine, metric hp				Electric motor, kW				
	Up to 1	Up to 2	Up to 5	Over 5	Up to 2	Up to 6	Up to 10	Up to 20	Over 20
Addition to calculated power, %	100	50	20	20 to 10	100	50	25	15	10

The operational economy of a pumping set (pump and motive unit) is expressed by the efficiency of the pump installation:

$$\eta_{pi} = \eta_m \eta_{op}$$

where η_m = efficiency of the motive unit driving the pump.

The efficiency of motive units employed in pump installations varies within the following ranges:

for electric motors η_m = 0.8 to 0.95
for steam engines η_m = 0.6 to 0.85.

1-10. Determining the Principal Dimensions of the Pump Cylinder and the Diameter of its Connections

The actual capacity of a pump is

$$Q = 60 \eta_v k \frac{\pi D^2}{4} Sn \text{ cu m per hour} \qquad (98)$$

Then the diameter of the liquid cylinder is

$$D = \sqrt{\frac{4Q}{60 \eta_v k \pi S n}} = 0.1455 \sqrt{\frac{Q}{k S n \eta_v}} \text{ m} \qquad (99)$$

If it is undesirable to assign the length of the piston stroke, its ratio to the cylinder diameter can be denoted as $\psi = \frac{S}{D}$ whence $S = \psi D$ and then equation (98) can be rewritten as

Then
$$Q = 60 \eta_v k \frac{\pi D^3}{4} \psi n \text{ cu m per hour}$$

$$D = \sqrt[3]{\frac{4Q}{60 \eta_v k \pi \psi n}} = 0.2765 \sqrt[3]{\frac{Q}{k \psi n \eta_v}} \text{ m} \qquad (100)$$

The piston stroke-to-cylinder diameter ratio and the number of double strokes per minute for pumps handling water can be taken from Table 15.

Table 15

Type of pump	n, rpm	$\psi = \frac{S}{D}$
Low-speed power pumps	from 40 to 80	2.5 to 2.0
Moderate-speed power pumps	„ 80 „ 150	2.0 to 1.2
High-speed power pumps	„ 150 „ 350	1.2 to 0.5
Extra-high-speed power pumps	„ 350 „ 750	0.5 to 0.2
Direct-acting pumps	„ 25 „ 130	1.75 to 1.0
Hand pumps	„ 20 „ 45	2.0 to 0.8
Hand fire pumps	„ 30 „ 60	3.5 to 2.5

The pump shaft speed n (number of double strokes per minute) for handling viscous liquids and ensuring normal suction of water pumps employed for this purpose can be selected by using the chart in Fig. 39.

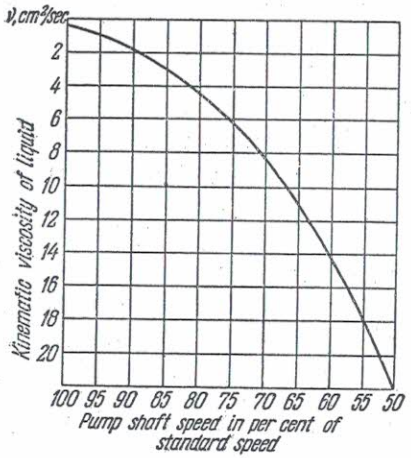


Fig. 39.

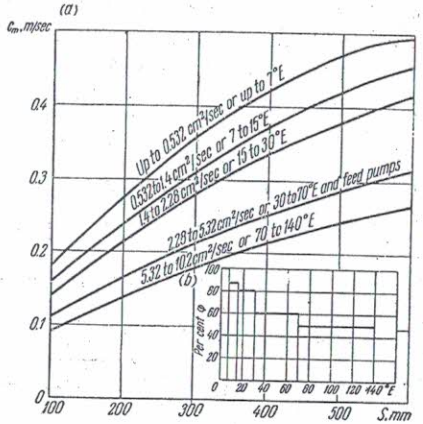


Fig. 40.

It may be noted that it is sometimes more convenient to calculate the cylinder diameter on the basis of the piston velocity. Since $Sn = 30c_m$, substituting into equation (99) will result in

$$D = \sqrt{\frac{4Q}{60\eta_v k \pi 30 c_m}} = 0.0266 \sqrt{\frac{Q}{k c_m \eta_v}} \text{ m} \qquad (101)$$

The maximum piston velocity of boiler feed pumps is about 0.3 m per sec; velocities for other types of pumps are listed in Table 12.

Among the values in the formulas used to determine the cylinder diameter, the pump capacity and the discharge stroke coefficient k are usually specified in the design assignment. The following discharge stroke coefficients have been established in practice for general service power pumps of various capacities:

simplex single-acting ($k = 1$) . . . . . 0.6 to 7.2 cu m per hour
simplex double-acting ($k = 2$) . . . . . 6 to 72 cu m per hour
triplex single-acting ($k = 3$) . . . . . 30 to 180 cu m per hour
duplex double-acting ($k = 4$) . . . . . 90 to 360 cu m per hour and more

Simplex and duplex direct-acting pumps are usually of the double-acting type ($k=2$ and 4, respectively) and their capacities vary in approximately the same ranges as those of power pumps.

The approximate piston velocities of direct-acting pumps are given in Fig. 40*a* as a function of the viscosity of the liquid being pumped and the piston stroke.

If viscous liquids are to be handled by pumps designed for water, the piston velocity should be reduced. Approximate piston velocities for such cases, expressed in per cent of φ, the velocity used to pump water, are given in Fig. 40*b*.

The velocity of the liquid flowing in the pump connections of the liquid cylinder differs from the piston velocity.

In shipbuilding practice the water flow velocity through inlet connections is taken to be in the range from 0.75 to 2 m per sec, and through discharge connections from 1.5 to 2.5 m per sec. Suction connection flow velocities of 0.6 to 0.7 m per sec and discharge connection flow velocities of 1 to 1.5 m per sec are suitable for handling viscous liquids.

If v_c is the liquid flow velocity through a connection then the inside diameter d_c of the connection may be determined by applying the equation of continuity. Thus

$$\frac{Q}{60 \times 60} = \frac{\pi d_c^2}{4} v_c$$

whence

$$d_c = \sqrt{\frac{Q}{3{,}600 \frac{\pi v_c}{4}}} = \sqrt{\frac{Q}{2{,}825 v_c}} = 0.0189 \sqrt{\frac{Q}{v_c}} \text{ m} \qquad (102)$$

Next we shall determine the principal dimensions of the steam cylinders of direct-acting pumps.

The forces of the pressure acting on the pistons of the steam and liquid cylinders of direct-acting pumps in steady-state operation differ only by the sum of the friction forces in the stuffing boxes, pivots, etc.

Let us employ the following notation:

D_s = diameter of the steam cylinder, m
p_{ls} = absolute pressure of the live steam when admitted into the steam cylinder (up to the valve chests), kg per sq m
p_{ss} = absolute pressure of the spent steam exhausted from the steam cylinder, kg per sq m; p_{ss} is taken equal to:
(*a*) 0.2—0.3 kg per sq cm, if the spent steam is exhausted into a condenser
(*b*) 1.15 kg per sq cm, if exhaust is into the atmosphere
(*c*) 1.2—3 kg per sq cm, if the spent steam is used to preheat the boiler feed water.

α_1=0.9 to 0.95 = wiredrawing factor in steam admission, specifying the loss in pressure in overcoming resistances in the valve chest

α_2=1.05 to 1.00 = wiredrawing factor in steam exhaust

α_i=0.9 to 0.95 = indicator diagram factor

φ_s=0.98 to 0.99 = factor accounting for the reduction in steam piston area due to the rod

φ_l=0.95 to 0.99 = factor taking into account the reduction in liquid piston area

$\eta_f=\eta_m\eta_h$ = factor taking into account the hydraulic and friction losses in the pump (Table 16)

p_s and p_d = mean absolute pressures, respectively, during the suction and discharge strokes in the liquid cylinder, kg per sq m (abs).

The following equation is obviously true for steady operation of the pump:

$$\varphi_s\eta_f\frac{\pi D_s^2}{4}(\alpha_1 p_{ts}-\alpha_2 p_{ss})\alpha_i=\frac{\pi D^2}{4}(p_d-p_s)\varphi_l \qquad (103)$$

where $(\alpha_1 p_{ts}-\alpha_2 p_{ss})\alpha_i=p_i$ = mean indicated pressure in the steam cylinder, kg per sq m.

Table 16

Piston stroke S, mm	Type of pump: General-service simplex and duplex — Number of double strokes per min	General-service simplex and duplex — Mean piston velocity, m per sec	Boiler-feed and for high-viscosity liquids — Number of double strokes per min	Boiler-feed and for high-viscosity liquids — Mean piston velocity, m per sec	Product of $\eta_m\eta_h=\eta_f$, % — Pressure, kg per sq cm (gauge): 10	20	70	200
75	80	0.2	40	0.10	50	47	45	39
100	75	0.25	38	0.125	55	52	50	43
125	72	0,3	36	0.15	60	57	54	47
150	65	0.325	33	0.17	65	61	58	51
200	56	0.375	28	0.19	70	66	63	55
250	48	0.4	24	0.2	75	71	67	58
300	45	0.45	23	0.23	78	74	70	60
375	40	0.5	20	0.25	80	76	72	62
450	37	0.55	18	0.27	82	78	74	64
600	30	0.6	15	0.30	85	81	77	65

(also for viscous liquids)

Note. The number of strokes indicated for duplex pumps refers to one cylinder, therefore it is necessary to multiply the tabular value by 2 to obtain the actual number of double strokes per minute.

Solving equation (103) for the diameter of the steam cylinder

$$D_s=D\sqrt{\frac{(p_d-p_s)\varphi_l}{p_i\varphi_s\eta_f}}\ \text{m} \qquad (104)$$

On pumps operating with expansion of the steam, factor α_1 depends upon the degree of admission δ and the diagram factor. Approximate values are listed in Table 17.

Table 17

δ	1	0.8	0.7	0.6	0.5	0.4
α_1	1	0.94	0.89	0.84	0.77	0.69

The diameter of steam engine cylinders for driving power pumps is determined by methods covered in the course on steam engines. This is outside the scope of the present textbook.

1-11. Valve Chests and Valves

Valve chests serve to connect the ends of the liquid cylinders of the pump by means of the valves to the suction and discharge lines at the corresponding positions of the piston.

Valve chests should: (1) be compact, (2) be arranged so that they do not increase the overall size of the pump, (3) have devices for adjusting the lift of the valves, (4) present the least possible hydraulic resistance to the flow of liquid in them, and (5) provide easy access to the valves for inspection, replacement and repair of the valve parts.

In accordance with the number of valves they contain, valve chests are classified as single-, double-, triple-, four- and multiple-valve types.

Valve chests may be simple or combined depending upon the number of inlet and outlet connections they have. Simple valve chests are usually connected to either the suction or discharge cavity while combined valve chests are connected to both cavities of the liquid cylinder.

Valve chests are also classified according to the number of chambers they contain as single-, double-, triple-, four- and multiple-chamber chests.

The purpose of the valves is to connect the ends of the cylinder or to disconnect them from the suction or discharge cavities of the valve chest at the proper times.

Valves may be of either the positive-action or self-acting type. The former are employed only on special-purpose pumps and are

actuated by valve gear whose operation is coordinated with piston travel. Cocks or common slide valves are frequently used in place of positive-action valves.

Self-acting valves operate automatically at the corresponding positions of the piston as they are opened and closed by the pressure of the liquid being pumped.

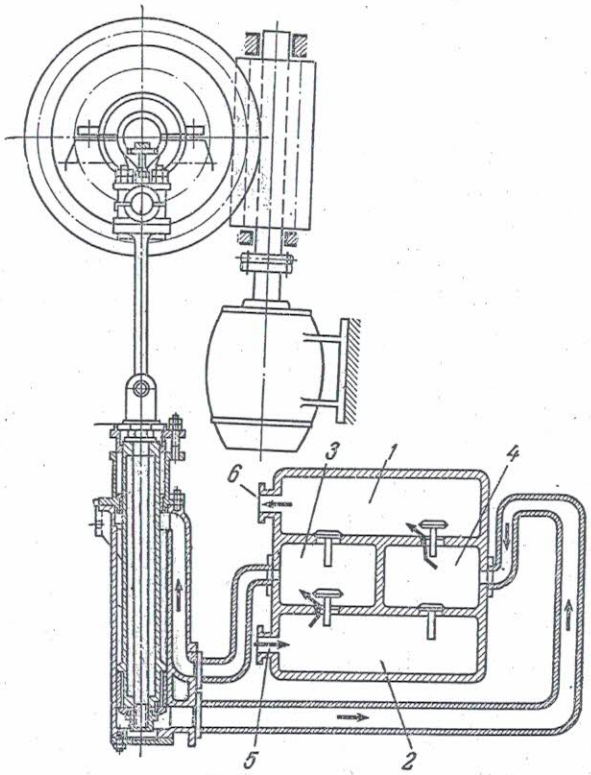


Fig. 41.

Simplicity in design and dependability in operation have found self-acting valves wide application. The pump valves are housed in the valve chests which adjoin the cylinders.

The arrangement of the valves in the valve chest of a double-acting pump is shown schematically in Fig. 41. The chest consists of a common discharge chamber *1*, suction chamber *2* and separate chambers *3* and *4* for the upper and lower ends of the pump cylinder, respectively. Liquid is admitted into the valve chest through flange

connection *5* and discharged through flange connection *6*. At the moment illustrated in Fig. 41, the discharge stroke is ending in the lower end of the cylinder and the suction stroke in the upper end.

The construction, dimensions, arrangement and material of the valves must ensure:

(1) tightness of closed valves;

(2) lifting and seating of valves without knocking;

(3) rapid lifting and seating of valves;

(4) low losses of head when the liquid passes through the valve or lifts it from its seat;

(5) resistance to corrosion by the liquid being handled.

Accurate fitting of the valve parts and careful lapping of the working surfaces on the valve disk and seat are required to obtain tight valves which reduce volumetric losses, or slip.

Quiet operation of valves without knocking is achieved by proper valve design, minimum possible weight of the moving elements of the valve and the selection of a proper lift. Quiet operation reduces valve wear.

The following formula, used in checking whether power pumps handling water will operate quietly, has been devised by Prof. I. Kukolevsky:

$$nh_{v\,max} \leqslant 600 \text{ to } 650 \qquad (105)$$

where h_{vmax} = maximum lift of the valve, mm
n = speed of the pump shaft, rpm.

If the lift needed to provide the required port opening is too high for one valve to achieve, the number of valves is increased so that their lift is in the range from 4 to 10 mm. In direct-acting steam pumps, the maximum valve lift may reach 30 mm because of the piston dwell at the end of the stroke. No formulas have been devised for determining at what lift and shaft speed knocking begins in direct-acting pumps. In this case, a check may be made on the basis of the valve closing velocity: $c_v < 60$ to 65 mm per sec.

Sometimes the working surfaces of valves are faced with leather, rubber, lignum vitae or plastics to achieve quiet operation.

Knocking of the pump valves leads to rapid wear of the lapped surfaces of the valve and its seat and annoys the pump operating personnel. Knocking indicates the lack of a layer of liquid between the lapped surfaces so that the valve drops directly onto its seat at a high velocity. Rapid, premature wear of the valves is highly detrimental to pump performance since it lowers the volumetric efficiency. The requirement that valves open and close rapidly is based on an effort to increase volumetric efficiency and reduce the hydraulic losses of the valve. Such losses are sharply reduced when the valve

is wide open; this increases pump efficiency. Rapid valve opening and closing can also be facilitated by using a small lift which can be obtained by proper selection of the valve weight and strength of the closing spring. A reduction in the hydraulic losses will increase the useful head of the pump because the losses of head in the valves will be reduced. These losses depend upon the design of the valve, its weight, force of the closing spring and the liquid flow velocity in the valve port opening.

In addition to factors ensuring normal pump operation, the selection of the design and material of the valves should be based upon their cost of manufacture and wear resistance.

Pump valves may be:

(1) metallic, made of bronze or various grades of steel,

(2) nonmetallic, made of leather, rubber, plastics or other pliable materials,

(3) composite, which are metallic valves with a nonmetallic facing that provides tightness.

There is a very large diversity of chest and valve designs.

A single-chamber valve chest with flat disk inlet *11* and discharge *12* valves is shown in Fig. 18.

A single-chamber combined chest with bottom-guided wing valves for suction and discharge is shown in Fig. 16.

The direct-acting pump illustrated in Fig. 19 has a four-chamber combined chest with inlet and discharge valves.

The pump design shown schematically in Fig. 41 incorporates a double-chamber combined valve chest.

A compact design of a pump may be achieved by employing combined valve chests but the losses of head in them are larger than in the simple type.

Metallic valves are used almost exclusively in low-, medium- and high-pressure pumps. The types of valves are illustrated in Table 20.

Conical disk valves offer less hydraulic resistance but are more difficult to manufacture than the flat type. Even the slightest misalignment of the axes of the valve and its seat, or difference in their tapers will lead to leakage through conical disk valves.

Thin disk valves find application in pumps with a low discharge pressure since they have a lower hydraulic resistance when they lift from their seats and are simple in design. The disks *1* of such valves may be pressworked into a special shape (Fig. 42), consisting of one or several flat plates, or they may be of the ring type (Fig. 43) with thin flat rings *1*. The latter are installed to increase the port opening (slot between the disk and valve seat). If thin disk valves consist of several plates, sometimes holes are provided in each plate to prevent them from sticking together.

The lift of a valve is adjusted by regulating the retainer *2* along the stud.

Various pliable materials are used to make disks *1* of nonmetallic valves (Fig. 44). The kind of material to be employed depends upon the operating conditions, pressure created by the pump and the liquid being handled.

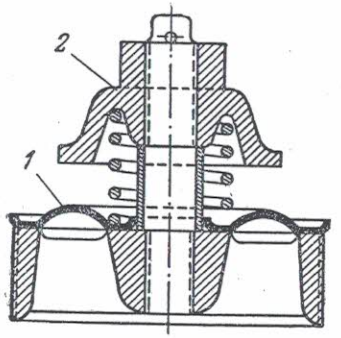


Fig. 42.

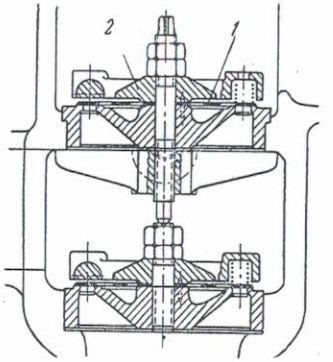


Fig. 43.

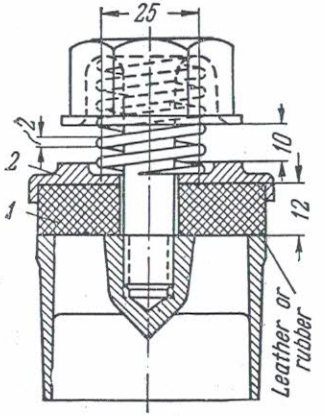


Fig. 44.

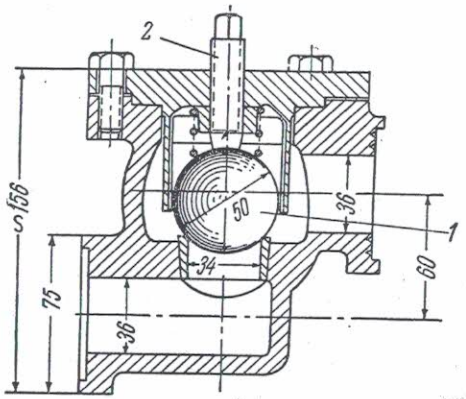


Fig. 45.

A nonmetallic valve is simple to manufacture; it requires no lapping in operation and is airtight.

The necessary rigidity may be imparted to the nonmetallic disk *1* by a metal reinforcing plate *2*. This is called a composite valve. Nonmetallic disks without reinforcement are also employed. Pumps

handling thick, polluted liquids are designed with ball valves (Fig. 45) in which a ball *f* is used as the closing element.

Multiple-ring valves are employed in high-capacity pumps to reduce the loss of head of the liquid in passing through the valve by increasing the port opening and thereby reducing the liquid flow velocity. A disadvantage of these valves is that the retainers and rings may become warped and lead to valve leakage. Liquid velocity may also be reduced by providing a group of identical valves installed on the same valve deck. An advantage of group valves is their interchangeability, while the disadvantage is that they increase the overall size of the valve chest.

The lift of the valves illustrated in Figs. 42, 43 and 45 is adjusted by setting the retainer, set screw *2* (Fig. 45) or nut (Fig. 44) at the required height.

1-12. Loss of Head in Valves

A liquid loses a part of its head in lifting a valve from its seat and in passing through the port opening. This is required to impart to the valve and the column of liquid above it the initial acceleration necessary to set them into motion.

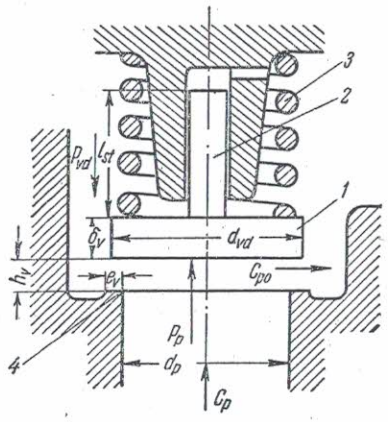


Fig. 46.

A valve can be lifted from its seat when the pressure p_p, in kg per sq m, that the liquid exerts on the lower surface f_p sq m (Fig. 46) of the valve disk is greater than the pressure p_{vd} kg per sq m exerted on the upper surface f_{vd} sq m of the disk *1* by the amount $h_{vo}\gamma$ kg per sq m which is the resistance of the valve to opening. Thus

$$p_p = p_{vd} + h_{vo}\gamma \text{ kg per sq m} \qquad (106)$$

Consequently, the loss in head in lifting the valve from its seat is

$$h_{vo} = \frac{p_p - p_{vd}}{\gamma} \text{ m } H_2O \qquad (107)$$

The pressure on the upper and lower surfaces of the valve disk can be determined from the equilibrium of forces acting upon the closed valve. In this position the force of the liquid pressure $p_p f_p$ kg acts on the underside of the valve disk. This force is balanced by the weights of the valve with its stem *2* and the spring *3* in the liquid being pumped, G_{vs} kg, the force of the initial load, R_s kg, on the spring, the force of the liquid pressure, $p_{vd}f_{vd}$ kg, on top of the disk and the inertia, I_v kg, of the valve directed downwards. Therefore,

$$p_p f_p = p_{vd} f_{vd} + G_{vs} + R_s + I_v \text{ kg} \qquad (108)$$

The resistance to opening offered by a discharge valve is determined by solving equation (108) for p_p and substituting its value into equation (107). Then, denoting the acceleration of the valve as j_v, we obtain

$$p_p = \frac{1}{f_p}(p_{vd}f_{vd} + G_{vs} + R_s + I_v) = \frac{1}{f_p}\left(p_{vd}f_{vd} + G_{vs} + R_s + G_{vs}\frac{j_v}{g}\right) \qquad (109)$$

and

$$h_{do} = \frac{p_p - p_{vd}}{\gamma} = \frac{\frac{1}{f_p}\left(p_{vd}f_{vd} + G_{vs} + R_s + G_{vs}\frac{j_v}{g}\right) - p_{vd}}{\gamma} =$$

$$= \frac{1}{\gamma f_p}\left[p_{vd}(f_{vd} - f_p) + (G_{vs} + R_s) + \frac{G_{vs}}{g} j_v\right] \text{ m}H_2O \qquad (110)$$

An analysis of equation (110) shows that the resistance offered by the valve to opening increases with the pressure above the disk, seating surface (contact area) of the disk, initial load on the spring, weight of the valve and its initial acceleration.

Equation (110) indicates that the valve should be designed as light as possible and the required load $(G_{vs} + R_s)$ on the valve should be provided by increasing the tension of the spring.

The pressure p_{vd} is determined from equation (55) which represents the pressure head acting upon the piston end during the discharge stroke. It is equal to the pressure in the discharge line directly following the valve at the end of the discharge stroke when the valve has already closed:

$$p_{vd} = \left(\frac{p_o}{\gamma} + z_d + \frac{L_d}{g} r\omega^2\right)\gamma \text{ kg per sq m} \qquad (111)$$

where L_d = referred length of the pipeline from the discharge valve to the outlet hole in the tank, in m.

In determining the loss of head required to lift the inlet valve from its seat it is necessary to find the pressure above the valve from equation (108) and to substitute it into equation (107). Thus

$$p_{vd}=\frac{1}{\gamma_{vd}}\left(p_p f_p-G_{vs}-R_s-G_{vs}\frac{j_v}{g}\right)$$

$$h_{so}=\frac{p_p-\frac{1}{f_{vd}}\left(p_p f_p-G_{vs}-R_s-\frac{G_{vs}}{g}j_v\right)}{\gamma}=$$

$$=\frac{1}{\gamma f_{vd}}\left[p_p(f_{vd}-f_p)+(G_{vs}+R_s)+\frac{G_{vs}}{g}j_v\right] \text{ kg per sq m} \quad (112)$$

The pressure p_p under the valve is determined from equation (48) for the piston position at the beginning of the suction stroke before the inlet valve opens:

$$p_p=p_a-\left(z_s+\frac{L_s}{g}r\omega^2\right)\gamma \text{ kg per sq m} \quad (113)$$

where L_s = referred length of the suction pipeline from the surface level of the liquid being pumped to the inlet valve.

The following data can be used to calculate the values of the terms in equations (110) and (112).

The diameter d_p of the port in the valve seat, required for a liquid flow velocity of c_p when there are z_v valves, is

$$d_p=\sqrt{\frac{4Q_s}{\pi c_p z_v}} \text{ m} \quad (114)$$

where c_p= 2 to 4 = permissible flow velocity for discharge valves, m per sec

c_p=0.75 to 2 = permissible flow velocity for inlet valves, m per sec

z_v=1 to 8 = number of valves in one valve deck.

Depending upon the pump capacity and the number of valves, the valve port diameter d_p ranges from 20 to 60 mm.

Other dimensions of the valves can be found from the following equations on the basis of the assigned value of d_p in cm:

maximum lift of the valve . . h_{vm} = (0.1 to 0.25) d_p cm

width of the lapped surface on metallic valves . . . . . e_v = (0.2 to 0.253) $\sqrt{d_p}$ cm

width of the lapped surface on nonmetallic valves or valves with a nonmetallic facing . . . . . e_v = (0.3 to 0.4) $\sqrt{d_p}$ cm

thickness of the valve disk . . δ_v = (0.1 to 0.2) d_p cm

height of the guiding stem or wings . . . . . l_w = (0.7 to 0.8) d_p cm

number of wings . . . . . i = 3 or 4

thickness of the wings . . . . . δ_w = 0.1 d_p cm

Strength calculations are to be made to check all the design dimensions of the valve.

In some cases it may be necessary to assign dimensions to the valve elements that are not within the above ranges.

Narrow contact (seating) surfaces (width—e_v) reduce the resistance to lifting the valve from its seat and ensure tighter contact between the lapped surfaces of the valve and seat, thereby preventing leakage of the valve. Therefore, the width of the seating surfaces can be taken less than that obtained by the above equations on condition that the pressure between the valve and its seat does not exceed the values given in Table 18.

Table 18

Material	Permissible pressure, kg/cm²
Wood	15 to 25
Rubber	15 to 25
Leather	40 to 60
Cast iron	130 to 140
Bronze	140 to 200
Phosphor bronze	200 to 600
Stainless steel	300 to 800

As a rule, the width of the seating (lapped) surfaces of the valves ranges from 2 to 6 mm while the maximum lift does not exceed 10 mm in power pumps or 30 mm in direct-acting pumps.

The weights of the valve and the spring are accurately computed from the dimensions of their working drawings.

As a tentative estimate the weight of the valve with the spring in air may be calculated from the formula

$$G_{vsa}=G_{va}+G_{sa}=0.001\gamma_v d_p^2 a(1+bd_p)+G_{sa} \text{ kg} \quad (115)$$

where G_{va} = weight of the valve in air, kg

G_{sa} = weight of the spring in air, kg

d_p = diameter of the port in the valve seat, cm

a and b = factors depending upon the design of the valve (Table 19)

γ_v = density of the valve material, kg per cu m.

Table 19

Type of valve	Flat disk valves		Conical-face valves	
	a	*b*	*a*	*b*
With upper guide stem	0.107	1.211 to 1.367	0.0284	4.56 to 5.14
With three unchamfered lower wings	0.107	1.981 to 2.122	0.0284	7.45 to 8.00
With four unchamfered lower wings	0.107	2.308 to 2.496	0.0284	8.7 to 9.37
With three chamfered lower wings	0.107	1.499 to 1.561	0.0284	5.64 to 5.88
With four chamfered lower wings	0.107	1.666 to 1.748	0.0284	6.26 to 6.56
With upper guide stem and three unchamfered lower wings	0.107	2.192 to 2.489	0.0284	8.24 to 9.00
Ditto but with four unchamfered lower wings	0.107	2.519 to 2.863	0.0284	9.85 to 10.87
Ditto but with three chamfered lower wings	0.107	1.7 to 1.928	0.0284	6.44 to 7.15
Ditto but with four chamfered lower wings	0.107	1.877 to 2.115	0.0284	7.05 to 7.95

Notes. 1. Valves are sometimes designed with upper wings; their volume is computed from the same data as for valves with lower wings.
2. The valves are shown schematically in Table 20.

The weight of the valve and spring in the liquid will be

$$G_{vs} = G_{vsa}\frac{\gamma_v - \gamma}{\gamma_v} \text{ kg}$$

The initial compression force R_{sp} of the spring ensuring airtight contact of the lapped surfaces when the valve is closed is

$$R_{sp} = y_0 k_{sp} \approx (0.667 \text{ to } 0.769) R_0 \text{ kg} \qquad (121)$$

where y_0 = initial deflection of the spring, cm
k_{sp} = load factor of the spring, equal to the load in kg, per cm of deflection
R_0 = compression force of the spring when the valve is fully opened, kg.

The compression force R_0 is found from the conditions of equilibrium of all forces applied to the spring at any open position of the valve. Thus

$$P_0 = R_0 + G_{vsa} \qquad (122)$$

whence

$$R_0 = P_0 - G_{vsa} \text{ kg} \qquad (123)$$

where P_0 = force of the liquid pressure acting on the underside of the open valve (see Table 20)
R_0 = $R_{sp} + 0.1\, h_v k_{sp}$ = compression force of the spring, in kg, when the valve is lifted to the height h_v, in mm.

The load on the valve equal to $H_v = \frac{R_0 + G_{vsa}}{\gamma f_v}$ m H_2O is listed in Table 21 as a function of the head created by the pump.

According to the formulas for P_0 (Table 20), the force of the liquid pressure acting on the underside of the valve disk increases with an increase in velocity of the liquid approaching the valve port, and decreases with an increase in the lift of the valve. A variation in these values leads to a corresponding variation in the compression of the valve spring so that at any moment in its opening movement the valve is in a state of equilibrium or, as they say, "floats" in the liquid.

The force of inertia of the valve and spring is determined from the formula

$$I_v = \frac{G_{vs}}{g} j_v \qquad (124)$$

The acceleration of the valve at the moment it is lifted from its seat is equal to the acceleration of the liquid under the valve. Therefore, this acceleration can be found by differentiating the equation of continuity for the extreme positions of the piston, thus

$$Fc = f_p c_p z_v$$

$$\frac{dc_p}{dt} = \frac{F}{z_v f_p}\frac{dc}{dt} = \frac{F}{z_v f_p} r\omega^2 \left(1 - \frac{x}{r}\right) = \frac{F}{z_v f_p} r\omega^2 = j_v \qquad (125)$$

Experimental data indicate that the hydraulic resistance of an open valve varies only slightly with the lift. The loss of head in an open valve can be computed, with sufficient accuracy for practical purposes, at maximum lift, i.e.,

$$h_{vl} = \zeta_v \frac{c_p^2}{2g} \qquad (126)$$

where ζ_v = coefficient of hydraulic resistance of the valve depending upon its design (Table 20).

In place of a single large valve it usually proves expedient to use several smaller, identical valves with the same total port area as the large valve. This reduces the loss of head in the valves and enables the lift of the valves to be decreased. Less lift ensures a lower

Table 20

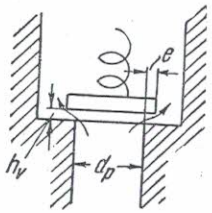
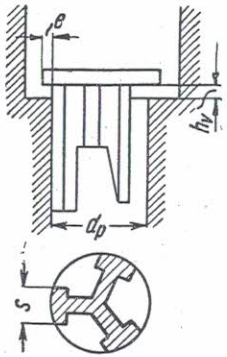
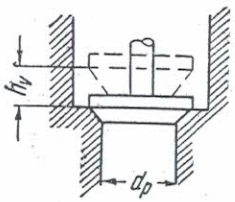
No.	Type of valve	Formulas and tables for determining the coefficient of resistance, ζ_v	Formulas for the force, P_0, of the liquid pressure acting on the valve from underneath, kg
1	Flat disk valve without wings (Fig. 47) 	$\zeta_v = 0.75 + 0.155\left(\frac{d_p}{h_v}\right)^2$ $\frac{d_p}{h_v}$: 10 | 8 | 6 | 4 ζ_v: 16.3 | 10.7 | 6.4 | 3.2	$P_0 = \gamma f_p \frac{c_p^2}{2g}\left[2.5 + 19\frac{e-0.1d_p}{d_p} + \left(\frac{d_p}{2.45h_v}\right)^2\right]$ (116)
2	Flat disk valve with wings (Fig. 48) 	$\zeta_v = 1.35 + 0.21\left(\frac{d_p}{h_v}\right)^2$ $\frac{d_p}{h_v}$: 10 | 8 | 6 | 4 ζ_v: 22.4 | 14.9 | 9.0 | 4.8	$P_0 = \gamma f_p \frac{c_p^2}{2g}\left\{2.25 + 17.1\frac{e-0.1d_p}{d_p} + \left[\frac{f_p}{0.54(\pi d_p - iS)h_v}\right]^2\right\}$ (117) where i = number of wings S = width of the wings
3	Conical-face valve with upper guide stem (Fig. 49) 	$\zeta_v = 2.6 - 0.8\left(\frac{d_p}{h_v}\right) + 0.14\left(\frac{d_p}{h_v}\right)^2$ $\frac{d_p}{h_v}$: 10 | 8 | 6 | 4 ζ_v: 8.6 | 5.2 | 2.8 | 1.6	$P_0 = \gamma f_p \frac{c_p^2}{2g}\left[-1.05 + \left(\frac{d_p}{3.56h_v}\right)^2\right]$ (118)

Table 20 (continued)

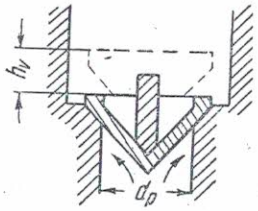
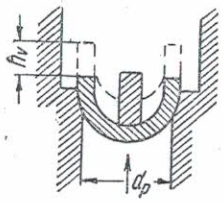
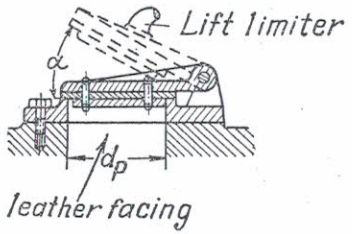
No.	Type of valve	Formulas and tables for determining the coefficient of resistance, ζ_v	Formulas for the force P_0 of the liquid pressure acting on the valve from underneath, kg
4	Conical valve (Fig. 50) 	$\zeta_v = 0.6 + 0.15\left(\frac{d_p}{h_v}\right)^2$ <table><tr><td>$\frac{d_p}{h_v}$</td><td>10</td><td>8</td><td>6</td><td>4</td></tr><tr><td>ζ_v</td><td>15.6</td><td>10.2</td><td>6.0</td><td>3.0</td></tr></table>	$P_0 = \gamma f_p \frac{c_p^2}{2g}\left[0.38 + \left(\frac{d_p}{2.72h_v}\right)^2\right]$ (119)
5	Spherical valve (Fig. 51) 	$\zeta_v = 2.7 - 0.8\left(\frac{d_p}{h_v}\right) + 0.14\left(\frac{d_p}{h_v}\right)^2$ <table><tr><td>$\frac{d_p}{h_v}$</td><td>10</td><td>8</td><td>6</td><td>4</td></tr><tr><td>ζ_v</td><td>8.7</td><td>5.3</td><td>2.9</td><td>1.7</td></tr></table>	$P_0 = \gamma f_p \frac{c_p^2}{2g}\left[0.96 + \left(\frac{d_p}{4.6}\right)^2\right]$ (120)
6	Flap valve (Fig. 52) 	<table><tr><td>Opening angle α°</td><td>15</td><td>20</td><td>30</td><td>40</td><td>50</td><td>60</td><td>70</td></tr><tr><td>Coefficient of resistance ζ_v</td><td>90</td><td>62</td><td>30</td><td>14</td><td>6.6</td><td>3.2</td><td>1.7</td></tr></table>	

Table 21

Pump	Head, mH_2O	Load, H_v, m liquid column
Low-pressure	Up to 50	0.4 to 0.6
Medium-pressure	50 to 500	1 to 2
High-pressure	Over 500	2 to 6

closing velocity of the valve which leads to silent operation and less wear of the lapped contact surfaces. Moreover, the interchangeability of valves in groups enables worn valves to be replaced more quickly.

1-13. Theory of Valve Motion

The law of valve motion in a power pump may be deduced by first determining the lift of the discharge disk valve upon steady operation of the pump (Fig. 46). As the valve is lifted with an instantaneous velocity of c_v, a space under the valve is freed of a volume equal to the area of the disk f_{vd} multiplied by the lift h_v. Before entering the pipeline beyond the valve, liquid, flowing from the pump cylinder at a velocity of c_p into the valve port with an area of f_p, first follows the valve, filling the space that it frees. Therefore, the volume of liquid delivered by the pump into the pipeline is less than the displacement of the piston by the volume swept through by the valve disk as it is lifted. When the valve is closed its disk displaces the liquid, remaining under the valve when it was opened, into the pipeline. Thus, the volume of liquid delivered beyond the valve as it closes is greater than the volume delivered by the piston to the valve by the volume swept through by the valve disk in closing. This phenomenon is known as the Westphal law.

According to this law the continuity equation for the discharge valves of a pump (Fig. 46) can be written as follows:

$$\mu\pi d_p h_v c_{po} z_v = Fc - f_{vd} c_v z_v \qquad (127)$$

where μ = discharge factor of the port opening (annular opening between the valve and seat when the valve is open)
c_{po} = theoretical velocity of the liquid in the port opening, m per sec
c_v = velocity with which the valve opens.

Assuming for simplicity that the pump has only one valve, we obtain

$$\mu\pi d_p h_v c_{po} = Fc - f_{vd} c_v \qquad (128)$$

Solving this equation for the lift h_v of the valve, we obtain

$$h_v = \frac{Fc - f_{vd} c_v}{\mu\pi d_p c_{po}} = \frac{1}{\mu\pi d_p c_{po}} (Fc - f_{vd} c_v) \qquad (129)$$

Assuming that the terms of the multiplier standing before the parentheses are constant values we can find the valve opening velocity:

$$c_v = \frac{dh_v}{dt} = \frac{1}{\mu\pi d_p c_{po}} \left(F \frac{dc}{dt} - f_{vd} \frac{dc_v}{dt} \right) \approx F \frac{dc}{dt} \frac{1}{\mu\pi d_p c_{po}} \qquad (130)$$

In practice, the term $f_{vd} \frac{dc_v}{dt}$ can be neglected without any appreciable error. Then, on the basis of equations (129) and (130),

$$h_v = \frac{1}{\mu\pi d_p c_{po}} \left(Fc - f_{vd} F \frac{dc}{dt} \frac{1}{\mu\pi d_p c_{po}} \right) \qquad (131)$$

but since

$$c = r\omega \sin\beta^\circ \quad \text{and} \quad \frac{dc}{dt} = r\omega^2 \cos\beta^\circ$$

then

$$h_v = \frac{1}{\mu\pi d_p c_{po}} \left(Fr\omega \sin\beta^\circ - f_{vd} F \frac{r\omega^2 \cos\beta^\circ}{\mu\pi d_p c_{po}} \right) \qquad (132)$$

Equation (132) enables the lift of the valve to be determined for any position of the piston as expressed by the angle β° of pump crankpin rotation. Moreover, the first term within the parentheses represents the variation in the instantaneous delivery of the pump depending upon the angle β° of crankpin rotation. The second term represents the instantaneous volume of liquid remaining under the valve during its motion.

On the basis of equation (132), rewritten as

$$h_v = \frac{Fr\omega \sin\beta^\circ}{\mu\pi d_p c_{po}} - \frac{f_{vd} Fr\omega^2 \cos\beta^\circ}{(\mu\pi d_p c_{po})^2} \qquad (133)$$

we can plot the curve shown in Fig. 53 with the valve lifts h_v as ordinates and the angles β° of crankpin rotation as abscissas.

The variable value of the first term in equation (133) corresponds to the sine curve *I* while that of the second term corresponds to the cosine curve *II*. Adding the ordinates of the two curves at identical values of β° we can plot curve *III* which expresses the law of valve lift variation during one stroke of the piston.

The curve in Fig. 53 shows that the discharge valve begins to open when the crankpin has turned through an angle of θ_1° from its extreme position. This corresponds to the angle of retarded opening of the inlet valve. When the piston reaches its other extreme position, the valve will still be open and its lift will equal h_v' since there will still be liquid under the valve. This liquid is removed and the

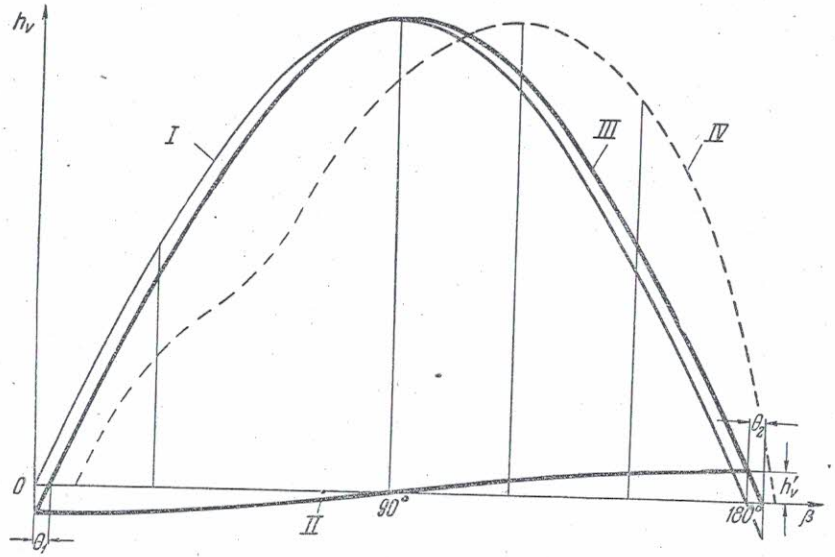


Fig. 53.

valve returns to its seat when the crankpin has rotated through an additional angle of θ_2° from its extreme position. The area bounded by the cosine curve and the axis of abscissas represents the volume swept through by the valve during its opening and closing movement. In the middle of the piston stroke, corresponding to an angle of crankpin rotation $\beta^\circ=90^\circ$, valve lift reaches its maximum value and is equal to $h_{vm}=\frac{Fr\omega}{\mu\pi d_p c_{po}}$. In another scale the ordinates of the sine curve are equal to the piston velocities and those of the cosine curve to the valve velocities at the corresponding moments of crankpin rotation. Thus, the valve velocity reaches its maximum value at the extreme positions of the piston and is equal to zero at the middle of the stroke. Since we know that the valve returns to its seat at an angle β° of crankpin rotation equal to $180^\circ+\Theta_2^\circ$, we can determine the angle of closing retardation, θ_2°. According to equation (133),

at the moment the valve returns to its seat

$$\frac{Fr\omega\sin(180^\circ+\theta_2^\circ)}{\mu\pi d_p c_{po}}-\frac{f_{vd}Fr\omega^2\cos(180^\circ+\theta_2^\circ)}{(\mu\pi d_p c_{po})^2}=0$$

$$\frac{Fr\omega(-\sin\theta_2^\circ)}{\mu\pi d_p c_{po}}=\frac{f_{vd}Fr\omega^2(-\cos\theta_2^\circ)}{(\mu\pi d_p c_{po})^2}$$

$$Fr\omega\frac{\sin\theta_2^\circ}{\cos\theta_2^\circ}=f_{vd}\frac{Fr\omega^2\dfrac{\cos\theta_2^\circ}{\cos\theta_2^\circ}}{\mu\pi d_p c_{po}}$$

$$\tan\theta_2^\circ=\frac{f_{vd}Fr\omega^2}{Fr\omega\mu\pi d_p c_{po}}=\frac{f_{vd}\omega}{\mu\pi d_p c_{po}} \qquad (134)$$

The angle of crankpin rotation θ_2° is usually less than 5° but it may be greater. Angle θ_1° varies within approximately the same limits.

The values of the discharge factor, μ, for water are listed in Table 22.

Table 22

h_v	0.1	0.2	0.3	0.4	0.5	0.6	0.8	1.0	1.5	2.0	2.5	3.0
μ	0.710	0.780	0.845	0.890	0.911	0.913	0.902	0.87	0.788	0.732	0.690	0.650
h_v	3.5	4.0	4.5	5.0	5.5	6.0	6.5	7.0	7.5	8.0	8.5	9.0
μ	0.622	0.599	0.578	0.560	0.545	0.532	0.523	0.515	0.507	0.500	0.493	0.485
h_v	9.5	10.0	11.0	12.0	13.0	14.0	15.0	16.0	17.0	18.0	—	—
μ	0.477	0.472	0.459	0.445	0.431	0.402	0.407	0.365	0.381	0.370	—	—

In the open position the valve is in a state of equilibrium in the flow of liquid passing through the port opening, due to the difference in pressure p_p-p_{vd} on the two sides of the valve. This difference in pressure is the loss in head, h_{vl}, in the open valve and is balanced by the load on the valve.

It follows from the above that

$$\frac{p_p-p_{vd}}{\gamma}=\frac{G_{vs}+R_0}{f_p\gamma}=H_v=\zeta_v\frac{c_p^2}{2g} \qquad (135)$$

Considering the port opening as a sort of nozzle and the flow through it as due to the difference in pressure, $p_p - p_{vd}$, we can write

$$c_{po} = \sqrt{\left(\frac{p_p}{\gamma} - \frac{p_{vd}}{\gamma}\right) 2g} = \sqrt{\frac{G_{vs} + R_0}{f_p \gamma} 2g} = \sqrt{2gH_v} = \sqrt{2gh_{vl}} = \\ = \sqrt{2g\zeta_v \frac{c_p^2}{2g}} = c_p \sqrt{\zeta_v} \qquad (136)$$

The actual velocity of the liquid passing through the port opening can be determined from the formula

$$c_{po.a} = \varphi_{po} \sqrt{2gH_v}$$

where $\varphi_{po} \approx 0.2$ to 0.25 = velocity factor.

The volume of liquid delivered by the piston under the valve at the moment the latter returns to its seat should be equal to the volume of liquid displaced by the valve in closing.

If c_0 is the piston velocity when the valve returns to its seat and $c_{v\theta}$ is the valve velocity at this moment then, according to the Westphal law,

$$Fc_0 - f_{vd} c_{v\theta} = 0$$

whence

$$c_{v\theta} = \frac{Fc_0}{f_{vd}} = -\frac{F}{f_{vd}} r\omega \sin(180° - \theta_2^\circ) = -\frac{F}{f_{vd}} r\omega \sin \theta_2^\circ \leqslant 60 \\ \text{to } 65 \text{ mm per sec} \qquad (137)$$

Therefore, the valve velocity in closing should be considered a negative value, and in lifting—a positive value.

The theoretical valve motion curve obtained above deviates from the actual curve because of the assumptions that were made, since it is impossible in an analytical deduction to take into consideration the effect of certain factors characterizing the lift of the valve. Therefore, the actual maximum lift of the valve will correspond to a moment where the piston will have passed the midpoint in its stroke. This is clearly shown by dashed line *IV* (Fig. 53) which represents the actual motion of the valve. The same line shows that the valve opens with a jerk due to the high resistance to opening at this moment. The inertia of the valve's mass and the fact that a higher velocity is imparted to it as it leaves its seat than the velocity corresponding to the given position of the piston cause the valve to lift higher than conditioned by the piston velocity. This leads to a reduction in valve lifting velocity upon further rotation of the crankpin

as shown by the flatter section of the curve which is followed by smooth lifting of the valve.

The velocity of the piston in direct-acting pumps is uniform over the greater part of the stroke and, therefore, the lift of the valve is constant over this part of the stroke. Besides this, the dwell of the piston at the extreme positions allows the valve to return to its seat when there is no flow of liquid through the valve. This is evident from Fig. 54 (section *ab*).

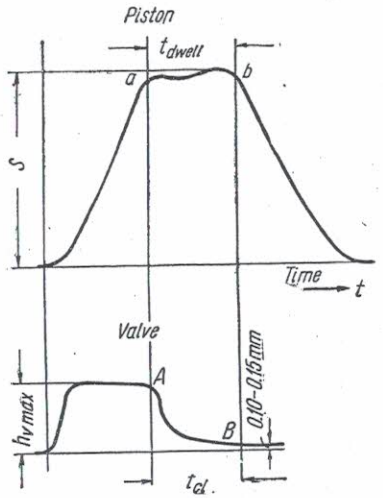


Fig. 54.

Using the notation of Fig. 54 and denoting the valve disk perimeter as l we can write the continuity equation for the closing of a valve in a direct-acting pump:

$$-f_{vd} \frac{dh_v}{dt_{cl}} = \mu l h_v c_{po}$$

$$-\frac{\pi d_{vd}^2}{4} \frac{dh_v}{dt_{cl}} = \mu \pi d_{vd} h_v c_{po}$$

$$dt_{cl} = \frac{d_{vd}}{4c_{po}\mu} \frac{dh_v}{h_v}$$

Integrating, we obtain the time required for the valve to close which is

$$t_{cl} = -\frac{d_{vd}}{4c_{po}\mu} \int \frac{dh_v}{h_v} = -\frac{d_{vd}}{4c_{po}\mu} \ln h_v + C_{int} \qquad (138)$$

The integration constant can be found if we take $t_{cl} = 0$ in equation (138). In this case $h_v = h_{vm}$ and then

$$C_{int} = \frac{d_{vd}}{4c_{po}\mu} \ln h_{vm}$$

Therefore, equation (138) can be rewritten as

$$t_{cl} = \frac{d_{vd}}{4c_{po}\mu} \ln \frac{h_{vm}}{h_v} \qquad (139)$$

Assuming in equation (138) that $h_v = 0.1$ to 0.15 mm we can obtain the time required for the valve to return to its seat:

$$t_{cl} = 0.5825 \frac{d_{vd}}{c_{po}\mu} \log \frac{h_{vm}}{h_v} \qquad (140)$$

1-14. Displacing Elements and Stuffing Boxes of Reciprocating Pumps

The displacing elements of reciprocating pumps are designed for creating a vacuum in the cylinder and for discharging the liquid from it. The displacing elements may be either pistons or plungers.

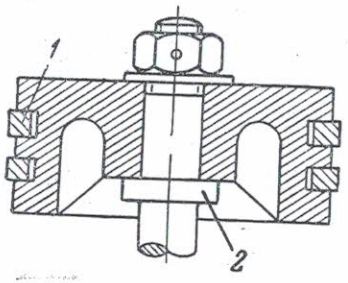


Fig. 55.

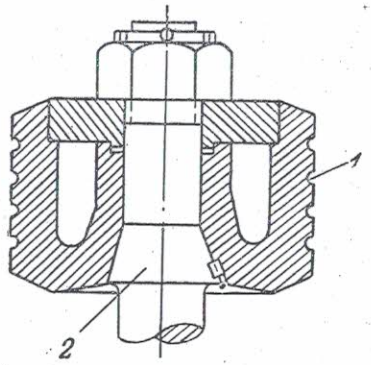


Fig. 56.

Pistons may be either of the solid or built-up type. Packing is provided to obtain airtightness between the cylinder walls and the cylindrical surface of the moving piston. The type of packing used depends upon the conditions under which the piston operates. In some cases, spring packing rings *1* (Fig. 55) are employed. They may be made of steel, cast iron, bronze, or various plastics. Other types of packing are labyrinth grooves (Fig. 56), compression packing and cup packing *1* (Fig. 57) of leather, rubber or rubberized fabric. Plain open pistons (Fig. 55) are usually used in single-acting pumps while built-up pistons (Figs. 56 and 57) are used in double-acting pumps. Piston rods sometimes have a taper *2* (Fig. 56) to ensure

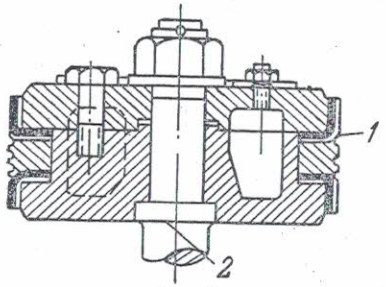


Fig. 57.

better alignment of the piston, a tight fit and easier disassembly. If the piston is fitted on the cylindrical part of the rod against a shoulder *2* (Figs. 55 and 57) it is more difficult to assemble and remove because of possible misalignment and jamming.

The friction surfaces of piston rings may be faced with white metal to facilitate the fitting of the rings and running in with the cylinder walls, as well as to facilitate repairs. Rings are often provided with labyrinth grooves to make them more airtight. The use of bucket pistons in which the inlet valve is mounted in the end of the piston enables the overall dimensions of the pump to be reduced.

The material of the packing depends upon the temperature of the liquid being pumped and its properties, the piston velocity and the pressure in the cylinder.

Any kind of packing can be employed when the temperature of the liquid is below 40°C; any kind with the exception of compression or cup packing can be used for temperatures in excess of 40°C.

All kinds of packing are suitable for piston velocities up to 1 m per sec; higher velocities exclude the use of compression and cup packing.

Any kind of packing may be used for cylinder pressures up to 5 atm; compression packing and labyrinth grooves cannot be used for pressures from 5 to 30 atm while only cup packing is suitable for pressures from 30 to 5,000 atm. The rim of a packing cup has a width from 4 to 35 mm, a thickness from 1.5 to 5 mm and the lip is bevelled at an angle of 20° to 30° towards the cylinder axis. The liquid pressure forces the lips against the wall of the cylinder. More pressure creates a better seal. The cylinder bore should be carefully machined with a smooth finish to prolong the service life of the cups, otherwise the cups will wear rapidly. Rapid cup wear may also be due to the pumping of liquids polluted with mechanical impurities.

High sealing capacity can be provided by cup and by ground and lapped packing, fair capacity (low leakage)—by compression or ring packing and low capacity (high leakage)—by labyrinth grooves.

The least friction and wear is observed in labyrinth grooves packing. Piston rings are subject to only slight friction and wear. High friction and wear is characteristic of cup packing.

A common drawback of pistons is the impossibility of checking the condition of the packing without opening the pump cylinder. Plungers do not have this disadvantage since there is no need for creating a seal between the surfaces of the plunger and cylinder.

Plungers of small diameter are cast of bronze, cast iron or steel (Fig. 58) while large diameter plungers are welded of pipe (Fig. 59). Plungers may be of various types depending upon whether they are single- or double-acting. Both open (Fig. 58) and closed (Fig. 59)

plungers are employed in single-acting pumps. Only the closed type is suitable for double-acting and differential pumps.

The ends of plungers used in high-speed pumps may be made with a conical or parabolic shape to reduce the resistance as they travel through the liquid. Steel plungers may have a sleeve *1* (Fig. 59) to protect them against corrosion and to facilitate their repair.

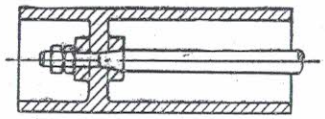


Fig. 58.

Stuffing boxes serve to seal plungers or piston rods where they extend from the cylinder. A stuffing box (Fig. 60) consists of the gland *1*, box proper *2*, packing *3* and the holding device. In design *I* the holding device consists of an eye bolt *4* with a nut *5* while in design *II* a removable bolt *6* with a nut *7* are used. The first design is more convenient in assembly and disassembly.

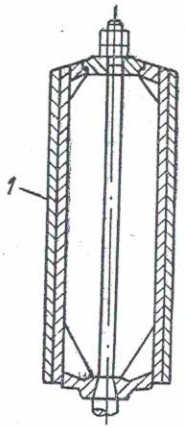


Fig. 59.

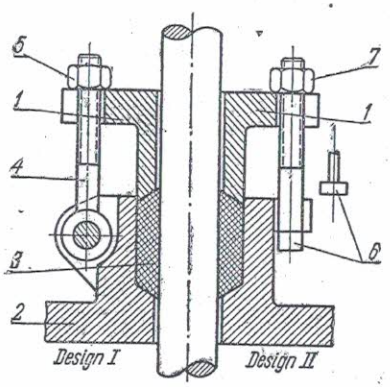


Fig. 60.

Packing may be of the nonmetallic, or soft, and metallic types.

Soft packings are employed for piston velocities from 2 to 3 m per sec, at temperatures of the liquid being handled up to 450°C and pressures up to 5 kg per sq cm.

Soft packing is made of cotton cloth, hemp or asbestos in the form of braided strands or rings (depending upon the operating conditions) impregnated with a graphite and grease mixture for lubricating purposes, as well in the form of cups made of leather, rubber or rubberized fabrics.

The elasticity of certain types of packing is increased by the provision of a rubber core *1* (Fig. 61*a*) surrounded by the main material

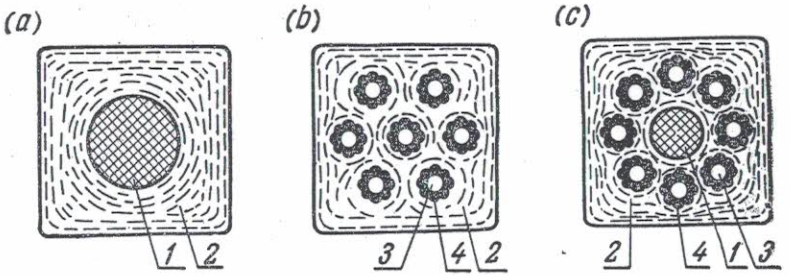


Fig. 61.

2 of the packing. In some packings the core *3* is of fibre surrounded by a bundle of lead wire *4* (Fig. 61*b*). Still another type is a combination of the first two types (Fig. 61*c*).

The packing is applied in the space (Fig. 60) formed by the stuffing box and the surface of the rod or plunger and is compressed by the gland *1* when nuts *5* or *7* are tightened. The bevelled end of the stuffing box bore and the bevelled face of the gland, in conjunction with the flexibility of the packing ensure that the packing tightly contacts the rod. No special facilities for supplying lubricant to the packing are provided for low piston velocities; either the lubricant impregnating the packing is considered sufficient or lubricant is applied in the clearance between the gland and the rod.

At high piston velocities (Fig. 62) the packing is divided into two parts *1* and *2* by a lantern ring *3* which has a hole *4* through which lubricant is applied directly to the rod. In the course of operation, the packing wears, begins to harden and loses its elasticity. The required seal is obtained by tightening nuts *7* of studs *6* to advance the gland *5*.

Two designs of screw type stuffing boxes, suitable for rods of small diameter, are illustrated in Fig. 63.

The drawback of design *I*, notwithstanding its simplicity, is that the internal thread in the stuffing box *2* impedes uniform movement of the packing when the threaded bushing *1* is advanced to compress it. This cannot ensure uniform pressure of the packing on the working surface of rod *4*. The introduction of a thrust ring *5* in design *II* eliminates this disadvantage.

Drawbacks of soft packing are their excessive wear, high friction losses and excessive length. A soft packing does not possess the capa-

bility of automatically improving the seal upon increase in pressure in the cylinder and therefore requires constant care and frequent replacement.

If a better seal is required in a shorter stuffing box at low piston velocities (up to 1 m per sec), the V-ring or U-cup type of packing is resorted to. One or several rings or cups are applied depending upon the head created by the pump.

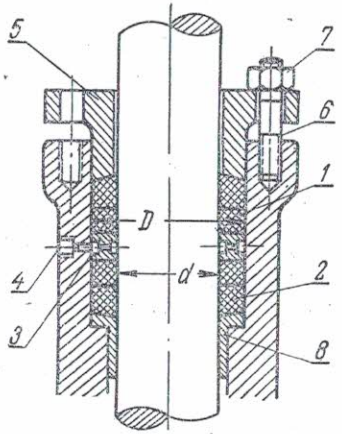


Fig. 62.

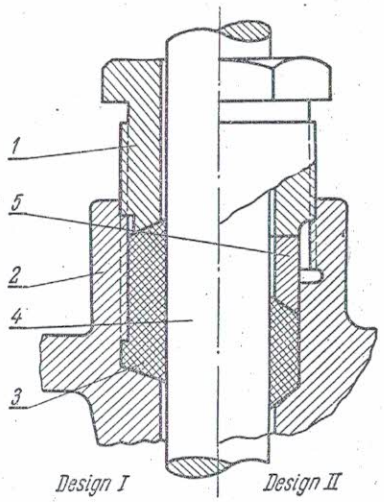


Fig. 63.

V-rings and U-cups withstand very high pressures and improve the seal automatically when the pressure is increased. They are made of leather, vulcanized rubber, cotton cloth, flexible plastics, or cotton cloth with a binding of vulcanized rubber. They are suitable for pumps handling liquids with a temperature up to 40°C or with temperature increases of short duration up to 70°C. V-rings and U-cups made of chrome-tanned leather can withstand liquid temperatures up to 85°C.

This type of packing is more wear-resistant and requires less attention and care in maintenance.

Fig. 64 illustrates a stuffing box packed with V-rings. One ring is sufficient for pressures up to 30 kg per sq cm; several may be applied for higher pressures. A disadvantage of this packing is that the lips wear more rapidly than the other surfaces of the rings.

This drawback has been eliminated in the U-cup packing shown in Fig. 65 in which the surface *1* of the thrust ring does not have the same shape as the cup. Another type of cup packing *1* made of rubberized fabric is shown in Fig. 66. Wedge rings *2* of white metal are inserted between the packing rings *1* in Fig. 67. After being run in, the white metal forms a continuous metallic surface.

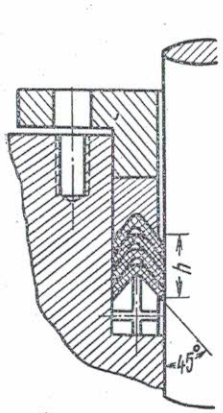


Fig. 64.

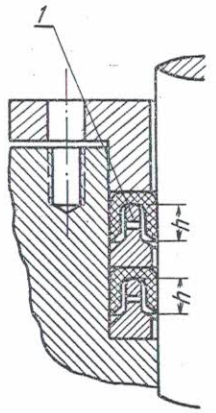


Fig. 65.

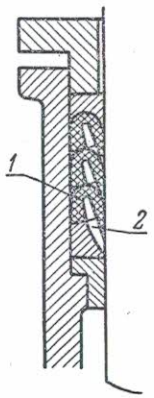


Fig. 66.

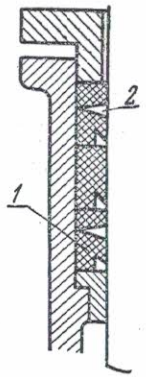


Fig. 67.

Semimetallic packing rings and metallic packing are employed for pumps with piston velocities from 3 to 14 m per sec, handling liquids at high temperatures and pressures.

A semimetallic packing (Fig. 68) is fitted into the stuffing box in the same manner as a soft packing and consists of a shell *1*, made of a soft alloy, and a soft flexible core *2* (usually of asbestos fibre with

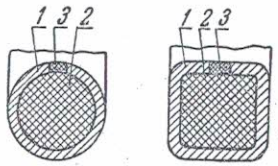


Fig. 68.

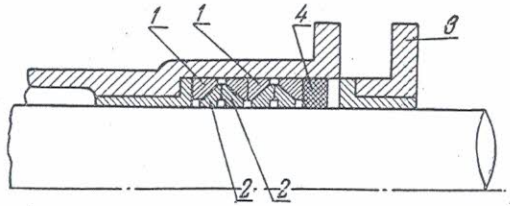


Fig. 69.

graphite) which is gradually squeezed out through hole *3* as the gland is tightened, and forms a reliable seal. A metallic packing (Fig. 69) consists of rings *2* with a triangular cross section made of a soft antifriction (bearing) alloy and cut into two half rings, alternating with split bronze rings *1*. The rings are forced against the internal surfaces of the stuffing box. The packing can be made more flexible by inserting a soft packing ring *4* before the gland. When pressure is exerted

by the gland *3*, the rings of bearing metal tightly contact the rod. Worn rings are replaced by new ones.

The least amount of friction and wear is observed in metallic packings; the greatest amount, in soft packings.

1-15. Reciprocating Pump Tests and Indicator Diagrams

An assembled pumping installation equipped with all instruments required for normal operation and testing is shown schematically in Fig. 70.

The suction line of the pump comprises the inlet box *1* with foot valves *2* and slit type strainer *3*, and the pipe with the necessary fittings.

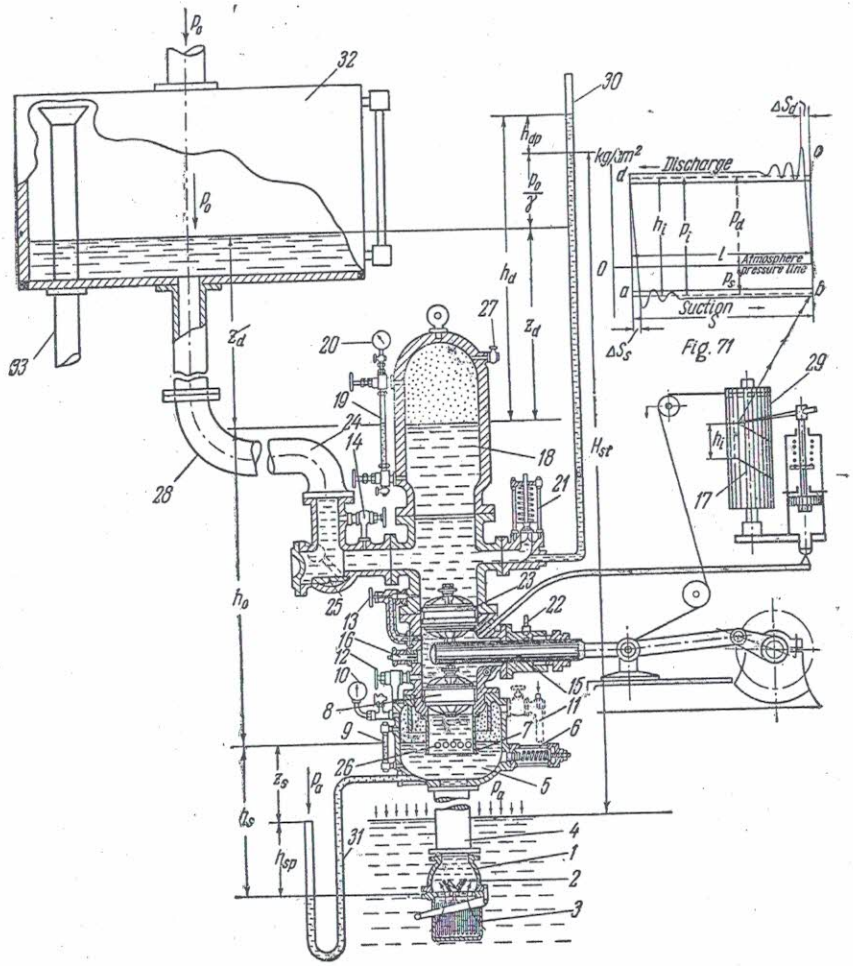
The foot valves *2* keep the suction line full of liquid even when the pump has been standing idle for some time. This ensures the drawing of liquid into the cylinder immediately after the pump is started. Strainer *3* excludes the possibility of large pieces of solid matter being drawn into the cylinder together with the liquid. The suction pipe *4* is connected to the vacuum chamber *5* which has a relief valve *6* to prevent the pressure in the chamber from rising above the permissible value. The inlet connection *7* links the vacuum chamber with the chest of the inlet valve *8* through which the liquid enters the pump cylinder.

Air is evolved from the water in the vacuum during the suction stroke and, since the water level drops during this period, the air may pass through the connection into the pump cylinder impairing normal operation. This is avoided by the provision of the small holes *26* in the suction connection which allow the air to enter the cylinder only in small portions. The water level in the chamber is checked by the gauge glass *9* while vacuum gauge *10* indicates the air pressure. Sometimes the vacuum chamber is equipped with an air pump *11* to exhaust the air from the chamber and to create a vacuum in it.

If the suction line is empty it can be filled with water from the discharge line *24* (this is called priming the pump) through the priming valves *14*, *13* and *12*.

The discharge valve *23* connects the cylinder, in which the plunger *15* reciprocates, to the air chamber *18* mounted in the discharge line *24*. The air chamber has a pressure gauge *20*, gauge glass *19* and air cock *27* for releasing excess air from the chamber whenever necessary. In high-pressure pumps, due to the absorption of the air by the liquid being pumped, cock *27* serves to supply the chamber with air at the required pressure. The pump cylinder has a snifting valve *16* for drawing air from the atmosphere into the cylinder from where it

is discharged into the air chamber. A relief valve *21* is provided to prevent the pressure in the discharge line from rising above the permissible value.



Figs. 70-71.

Flap valve *25* disconnects the pump from the discharge line. Oil for lubricating the plunger is supplied through tube *22*. Plunger *15* is driven by a crank mechanism as is the indicator drum *29* which serves to take indicator diagrams *17*. Whenever possible a manometer *30* is used to make pressure measurements due to its higher accuracy;

the vacuum is measured with a U-tube manometer *31* operating upon the same principle. During operation liquid is delivered along the discharge line *28* to tank *32*. Overflow drain *33* precludes overfilling the tank.

Pumps are subjected to mechanical and hydraulic tests.

Mechanical tests are conducted to determine and estimate the quality of pump assembly and installation, and the convenience of servicing the pump. This can be done by checking:

(a) for absence of abnormal knocking,
(b) for absence of overheating of parts subjected to friction,
(c) the airtightness of all joints,
(d) the operation of the relief valves,
(e) the lubricant consumption,
(f) the steam consumption,
(g) whether the design dimensions of the pump parts suit the purposes of the parts,
(h) the hydraulic resistance of the valves,
(i) the intensity of the vibration,
(j) the accessibility of the pump for inspection and for determining the temperature of parts which require periodic attention during operation,
(k) the convenience of operation of the pump.

The purpose of hydraulic tests is to determine the capacity of the pump, the pressure it creates, the power required to drive the pump, the discharge factor and the overall efficiency. Before beginning the tests it is necessary to measure the principal dimensions of the liquid cylinder and to check whether the instruments to be used in the tests are serviceable.

Pump capacity may be measured by various methods, for example, with a calibrated tank, Venturi and Woltman flow meters, weirs, Pitot tubes, orifice meters and diaphragms.

The pressure and vacuum created by the pump are measured by vacuum gauges and manometers of various types.

Vacuum gauges are installed in the suction line before the inlet valve; manometers in the discharge line after the discharge valve.

The indication of the vacuum gauge h_{vac} in mH_2O gives the value of head, h_{sp}, required to overcome all resistances in the suction line plus that required to raise the liquid in it to a height z_s. In the same units the manometer indicates the head, h_m, which is equal to the head h_{dp} lost in overcoming all resistances in the discharge line plus the static head, z_d, lost in raising the liquid to its point of destination and in overcoming the head, $\frac{p_0}{\gamma}$, in the discharge tank.

Neglecting the difference in the velocity heads in the suction and discharge lines and denoting the difference in height of the positions

of the vacuum and discharge manometers in metres as h_0, we obtain the total manometric head, or column, created by the pump (Fig. 70):

$$H = h_{vac} + h_m + h_0 = (h_{sp} + z_s) + (h_{dp} + z_d) + h_0 + \frac{p_0}{\gamma} =$$
$$= (h_{sp} + h_{dp}) + \left(z_s + z_d + h_0 + \frac{p_0}{\gamma}\right) = (h_{sp} + h_{dp}) + H_{st}$$

where $H_{st} = z_s + z_d + h_0 + \frac{p_0}{\gamma}$ = useful static head.

The capacity and head created by the pump being known, we can calculate the power required to overcome all resistances outside of the pump cylinder:

$$N_u = \frac{\gamma Q_s H}{75} \text{ metric hp} = \frac{\gamma Q_s H}{102} \text{ kW} \qquad (141)$$

The actual total power required to produce useful work and to overcome harmful hydraulic and mechanical resistances in the pump cylinder is determined by a dynamometer. The latter has devices for measuring the force T applied to the end of the dynamometer lever. Since the length L of the lever is known and the speed of the pump shaft can be measured with a tachometer, the torque on the motor shaft can be determined ($M_{sh} = TL$) and therefore the power developed on the shaft will be

$$N_m = \frac{TL\omega}{75} = \frac{M_{sh}\omega}{75} \text{ metric hp} \qquad (142)$$

Thus the pump efficiency will be

$$\eta_{op} = \frac{N_u}{N_m} \qquad (143)$$

The indicated power N_i of the liquid cylinder is calculated on the basis of the mean indicated pressure in the cylinder which is determined from the indicator diagram:

$$N_i = k \frac{p_i F_m S n}{75 \times 60} \text{ metric hp} \qquad (144)$$

where k = discharge stroke coefficient of the pump
F_m = mean area of the piston, sq m
for a simplex single-acting pump $F_m = F$
for a simplex double-acting pump $F_m = (F - \frac{f_{rod}}{2})$
for a triplex single-acting pump $F_m = F$
for a duplex double-acting pump $F_m = (2F - f_{rod})$
f_{rod} = cross-sectional area of the piston rod, sq m.

The mechanical efficiency of the liquid cylinder is

$$\eta_m = \frac{N_i}{N_m} \qquad (145)$$

The volumetric efficiency of a pump is the ratio of the actual capacity to the theoretical value:

$$\eta_v = \frac{Q}{Q_t} = \frac{Q}{60kF_mSn} \qquad (146)$$

The steam consumption required to drive the pump may be determined with a steam-flow meter or by weighing or measuring the condensed steam.

It is necessary to take indicator diagrams in the course of pump tests. The nature of abnormal operation of a pump and its causes can be determined by the shape of the diagrams. A normal indicator diagram (Fig. 71) gives an idea of the processes that take place during the working cycle of the pump. The suction stroke ends at point *b*. The inlet valve is closed by the liquid pressure produced in the discharge stroke. When this pressure reaches a value equal to the full discharge head, during which time the piston travels a distance ΔS_d, the discharge valve opens (point *c*) and liquid is delivered into the discharge line. The discharge stroke continues up to point *d*. At point *d* the piston begins its suction stroke. When it has travelled a distance ΔS_s the discharge valve closes, the pressure drops (line *da*) and liquid begins to enter the cylinder.

The wavy lines at the beginning of the suction and discharge strokes indicate the variation in the liquid pressure under the valve after the latter is lifted from its seat.

An analysis of the indicator diagrams taken (Fig. 72) and comparison with normal diagrams enable conclusions to be drawn concerning the defects in the operation of the pump.

Delayed closing of the inlet and suction valves is reflected in indicator diagram *1* by the more slanted lines *da* and *bc*. The discharge valve opens only after the piston has travelled a distance ΔS_d of the discharge stroke; at this time the inlet valve returns to its seat. The inlet valve opens only after the piston has travelled a distance of ΔS_s on the suction stroke. Thus, in the two strokes of the piston, the pump displacement is reduced by $F(\Delta S_d + \Delta S_s)$ cu m. This defect can be eliminated by increasing the load exerted by the spring on the valve.

An insufficient load on the inlet and discharge valves may lead to slip (leakage) through the valves at the beginning of the stroke. The valves seat properly only after the piston has travelled a part of the stroke. This is evident on indicator diagram *2* from the distances ΔS_d and ΔS_s. This trouble is remedied by increasing the compression of the valve springs.

In cases of leakage through the inlet and discharge valves, as well as through the rod packing due to poor fitting or wear, the pressure in the cylinder varies gradually after the valve is closed. This is evidenced by the rounded corners on indicator diagram *3*. This

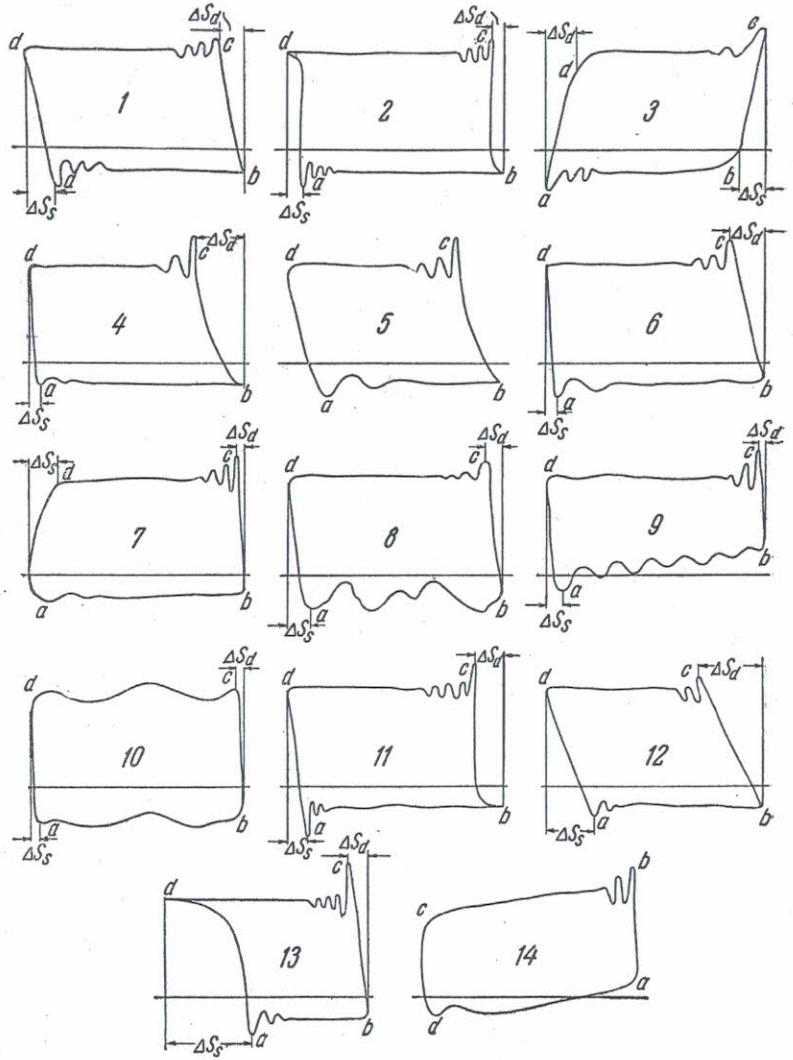


Fig. 72.

defect is remedied by cleaning or lapping the valves and by tightening the packing gland.

If air is drawn into the suction line and into the cylinder or if air and vapours are evolved from the liquid being handled, the air

and vapour occupying a part of the cylinder volume are compressed by the piston in the discharge stroke (along line *bc* of indicator diagram *4*) and are forced into the discharge line. The discharge valve will open only when the air and vapours of the liquid are compressed to the discharge pressure. The presence of air appreciably reduces the volumetric efficiency of the pump. Penetration of air into the suction line is eliminated by finding the leak and repairing the line; the evolution of vapours can be reduced by eliminating the causes of their formation.

If the air that enters the cylinder is not forced out during the discharge stroke and forms an air pocket due to poor arrangement of the discharge valve, the liquid will enter the cylinder during the suction stroke only after the air has expanded along the line *da* of indicator diagram *5* until it reaches the suction pressure. This also reduces the volumetric efficiency.

Indicator diagrams similar to *4* and *5* are also obtained in pumping hot liquids with a low vapourization point. In this case, the valves knock during operation.

When there is a large slip through the discharge or suction valve due to clogging of the valve, breakage of the disk or other causes, the discharge or suction pressure will be reached after the piston has travelled a certain distance on its stroke (ΔS_d on diagram *6* or ΔS_s on diagram *7*). The greater the clearance between the valve disk and its seat, the greater the slip of liquid back into the pipeline and, therefore, the flatter the lines *bc* and *da* of the diagrams *6* and *7*, respectively.

If air is drawn in slightly or if the liquid level in the vacuum chamber drops below the snifting valve, the line on indicator diagram *8* representing suction will be wavy. Indicator diagram *9* shows the operation of a negative-lift pumping installation. Here the line representing suction is above the atmospheric pressure line and large variations of the suction pressure are observed.

If the vacuum and air chambers are incorrectly installed or are too small, inertia forces will affect the operation of the pump. This is evident in indicator diagram *10* from the wavy lines for suction and discharge which resemble a harmonic curve.

The inlet valve may jam at the very beginning of its lift so that a higher vacuum will be required to open it than under normal conditions. This leads to an increase in the height of the peak *a* as in diagram *11*. Jamming of the same valve as it returns to its seat prevents the required pressure from being obtained until the valve seats properly. During this time the piston will have travelled a distance ΔS_d. The form of the indicator diagram for high-pressure pumps is affected by the resilience of the liquid remaining in the dead space. In indicator diagram *12* this is shown by the sloping lines *bc* and *da* which are the result of the elastic compression and expansion of the liquid.

If the discharge valve jams during opening, a higher pressure will be required to open it than in normal operation (point *c* in diagram *13*). On the other hand, if this valve jams during closing, pressure in the cylinder will not fall even though the suction stroke has begun. The valve returns to its seat only after the piston has travelled a distance ΔS_s on its stroke. Indicator diagram *14* is typical of pumps which are not equipped with vacuum or air chambers.

1-16. Operation of Reciprocating Pumps

All the stipulations of reciprocating pump operating rules and instructions of the pump manufacturer must be carefully observed. It is necessary before starting a pump to make sure by careful inspection that all joints are in proper condition and that all foreign objects hindering pump operation are removed. The latter can be checked, especially after repairs, by turning the pump shaft one or two revolutions by hand, after first opening the drain valves. Next, inspect and clean, if necessary, all lubricating devices and fill them with oil; apply oil from an oil can to the rods where they extend from the stuffing boxes and to all linkages that are not lubricated automatically. Open the valves and cocks of the piping leading to the measuring instruments. If the pump has been operating for some time without inspection of the packing, it should be inspected and replaced if excessive wear is observed. To ensure a proper seal, the half-rings of the packing should be arranged so that their butted joints are staggered on each ring at 90° in respect to the adjacent ring. The holding device is to be tightened so as to ensure uniform pressure on the packing and to avoid cocking of the gland. This is achieved by drawing up the nuts on the studs uniformly. Cooling water is supplied to the coil in the oil bath of the worm gearing. Then the inlet and discharge valves of the pump connections are opened, as well as all the necessary valves in the suction and discharge lines.

It is necessary each time the pump is started to check whether the relief valve opens properly.

Just before starting the pump the cocks in the lubricating lines are opened to supply oil to the friction surfaces. Having completed these preparatory operations, the motive unit may be started to drive the pump. If the speed of the motive unit is variable, it should be started at the lowest speed, which should be gradually increased to the normal running speed.

During pump operation:

(1) observe the indications of the instruments both for the liquid cylinder and for the motive unit,

(2) see that ample lubricant is supplied to the friction surfaces, fill the lubricating facilities periodically with lubricant,

(3) attend to the vacuum and air chambers, replenishing the lost compressed air in them by opening the air cock as required,

(4) check the oil level in the bath of the worm gearing, not allowing the level to fall lower than 40 to 55 mm from the lower end of the sight gauge, and see that the oil circulates properly in the bath,

(5) check the oil temperature in the bath, not allowing it to rise more than 45°C above the ambient temperature,

(6) check the temperature of friction parts by touch, not allowing it to rise above the stipulated values.

If signs of abnormal pump operation appear (overheating of friction parts, sharp knocks, etc.), try to find the cause and, if there is danger of a breakdown, stop the pump.

The most frequently occurring causes of poor pump operation are given in the following.

Failure of the pump to deliver liquid may be the result of one of the following causes:

(1) the foot valves are closed,

(2) leakage in the pipelines,

(3) the inlet strainers and filters are heavily clogged.

The reasons why a pump does not reach normal capacity or develop normal pressure may be:

(1) The valves in the discharge and suction lines are not fully opened.

(2) The filters, screens or foot valves of the suction inlet in the tank are clogged.

(3) There is air inflow through leaks in the joints of the suction line, through the gaskets between the joints of the pump parts (fittings, valve chest covers of the cylinders, decks of the inlet and discharge valves, etc.), through the air cocks, blow valves or pressure gauge cock. During pump operation air inflow can be detected (a) by testing the joints with a candle flame, and (b) by wetting the joints with a liquid (preferably a viscous liquid). In the first case the flame will be deflected towards a leaky joint while in the second a film of the liquid will be drawn into the leak. A hydraulic test may be applied to find the air inflow leak if the pump is standing idle.

(4) There is considerable bypassing of liquid from the discharge to the suction line along the bypass line through the relief valve due to leakage, breakage or incorrect adjustment of this valve (the spring is insufficiently compressed).

(5) The liquid being pumped leaks through the rod packing due to incorrect arrangement of the packing rings, excessive wear of the packing, or its poor or incorrect compression.

(6) There is leakage in the piston packing due to excessive wear, scoring of the cylinders, breakage of the packing rings or their incorrect arrangement.

(7) There is excessive slip of the liquid back into the cylinder or into the suction line due to broken (or leaky) discharge or inlet valves or due to hanging of the valves off their seats by foreign substances.

(8) The pump valve springs are either broken or poorly adjusted.

(9) The temperature of the water being pumped is raised or that of oil products is lowered. In such cases the capacity and pressure of the pump can be increased by reducing the pump speed.

Parts of the pump may overheat because insufficient oil is supplied to the friction surfaces, because dirty lubricant is used, because of too tight packing and also poor assembly of the pump.

The overheating of parts subject to friction can be eliminated by increasing the supply of lubricant to the friction surfaces. Another measure is to wash the bearings with kerosene or soluble oil emulsion and then to flood them with oil.

Overheating in the worm gearing may be the result of:

(1) insufficient oil in the bath or clogging of the oil circulation tubes,

(2) lack of cooling water in the oil-cooling coils of the bath or insufficient water supply because the cocks on the coils are not fully open,

(3) dirty oil in the bath,

(4) incorrect assembly (misalignment) of the gearing,

(5) an increase in speed of the motor shaft and, consequently, of the worm above the permissible value.

The bearings and the connecting rod ends may overheat because of:

(1) a lack of oil or wicks in the oilers, or incorrectly fitted wicks in wick lubrication,

(2) too tight cap bolts of the bearings,

(3) scoring in the bearing liners or on the shaft journals,

(4) insufficient supply of lubricant or its pollution,

(5) incorrect assembly of the bearings.

Knocking or pounding during pump operation may be due to:

(1) separation of the flow of liquid from the piston if the velocity of the latter is too high, leading to water hammer when liquid again runs up against the piston,

(2) excess air in the vacuum chamber and insufficient air in the air chamber, leading to water hammer,

(3) excess wear of the piston and rod packing leading to excess clearance and knocking,

(4) the presence of foreign matter in the cylinder,

(5) poor flexibility of the valve springs, broken springs, low load on valves, high valve lift or high piston velocities leading to knocking of the valves (if the valves knock as they open, increase the load; if they knock as they close, reduce the load),

(6) the fastening nuts or studs of the valves are loose,

(7) excessive wear of the bearing liners, crankpins, crosshead bushings, wrist pins or pistons, or loose bearing cap bolts.

To stop a reciprocating pump:

(1) Stop the pump motive unit.

(2) Close the valve on the suction and discharge connections of the pump.

(3) Close the cocks in the lubricating lines and in piping leading to the instruments, and pull out the wicks in the oilers.

(4) Close the valve in the line supplying cooling water to the coil in the worm gearing oil bath.

(5) Open the drainage valves.

(6) Wipe off the pump and inspect the fastening bolts and all joints.

In each separate case, the cause of poor operation of the pump must be discovered and eliminated.

Regularly, at least once every 500 to 1,000 hours of operation, the pump should be carefully inspected and, if required, the valves should be repaired or replaced, stuffing boxes should be packed and piston rings should be changed. After each 3,000 to 4,000 hours of service the pump should be completely disassembled to ascertain what repairs are necessary.

In addition to the above procedures, it is necessary to gradually warm up the steam cylinders in starting a direct-acting pump to a temperature approximately that of the live steam. This is done by gradually increasing steam admission to the cylinders with the drains open, this being necessary for uniform heating of all parts of the steam cylinder. The cylinder is sufficiently warmed when condensate ceases to run out of the drains in a full stream.

After this, the drains are closed and the steam throttle valve is opened gradually until the pump reaches its normal running speed, using the readings of the pressure and vacuum gauges as guidance.

If the pump is equipped with plain slide valves, operates with back pressure and the steam exhaust line is under pressure, the steam exhaust valve should be opened only after the live steam valve is open so that the steam from the exhaust line entering the valve chest does not push the valves away from their seats, bending the valve rods.

In stopping a pump which does not operate with back pressure, first the steam admission valves are closed, then the cylinder drains are opened and finally the steam exhaust valve is closed. If the pump operates with back pressure (with utilization of the spent steam), the steam exhaust valve is closed first to disconnect the pump from the spent steam line.

A power pump operating against a high head is started by opening the valve on the bypass line and with the valve in the discharge line closed. Under such conditions the pump operates within itself. This is done to avoid overloading the motive unit in the starting period. After the pump shaft has reached its normal running speed, the bypass valve is slowly closed, and the valve in the discharge line is opened simultaneously.

1-17. Steam Consumption and General Assessment of Reciprocating Pumps

The steam consumption of a reciprocating pump depends upon the design of the pump and the initial pressure of the steam (Fig. 73).

The consumption of saturated steam by direct-acting pumps running at rated speeds ranges from 30 to 70 kg per metric horse power hour. The effect of the initial condensation of the steam on its consumption is very great and, therefore, steam consumption increases with a decrease in piston velocity, reaching values of 130 kg per hp-h and even higher (Fig. 74).*

The steam consumption of direct-acting pumps can be determined by a modification of a method proposed by A. Voznesensky.

The mean indicated pressure in the steam cylinder (see Sec. 1-10) is

$$p_{si} = \alpha_i (\alpha_1 p_{ts} - \alpha_2 p_{ss}) \text{ kg per sq m} \qquad (147)$$

The required mean indicated pressure in the steam cylinder is found by writing the equation of the forces acting on the pistons of the steam and liquid cylinders:

$$p_{si}\varphi_s \frac{\pi D_s^2}{4} = R_{fr} + p_{li}\frac{\pi D^2}{4}\varphi_l = \frac{p_{li}\varphi_l}{\eta_m}\frac{\pi D^2}{4}$$

then

$$p_{si} = p_{li}\frac{\varphi_l}{\varphi_s}\left(\frac{D}{D_s}\right)^2 \frac{1}{\eta_m} \text{ kg per sq m} \qquad (148)$$

where $R_{fr} = (1.03 \text{ to } 1.1)\, k_s FSnp_{li}$ = work of the friction forces in the pump, kg-m

p_{li} = mean indicated pressure in the liquid cylinder, kg per sq m

η_m = mechanical efficiency of the pump.

* n_r and n_a are the rated and actual number of double strokes per minute.

Combining equations (147) and (148) we obtain the required pressure of the live steam before admission into the valve chest:

$$p_{ls} = \left[p_{li}\frac{\varphi_l}{\varphi_s}\left(\frac{D}{D_s}\right)^2 \frac{1}{\eta_m} + \alpha_i p_{ss}\alpha_2\right]\frac{1}{\alpha_i \alpha_1} \text{ kg per sq m} \qquad (149)$$

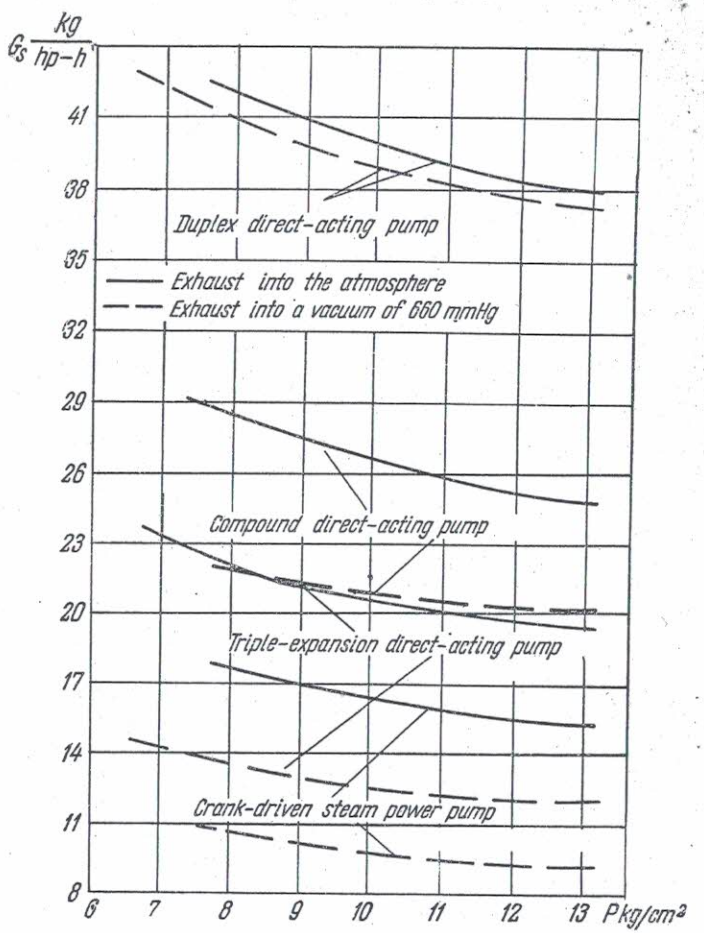


Fig. 73.

The hourly steam consumption for operation of the pump without steam expansion will be

$$G_{si} = 60k_s \frac{\pi D_s^2}{4} Sn\,\varphi_s \gamma_s \text{ kg per hour} \qquad (150)$$

where* $\gamma_s \approx 0.48\,p_{ls} + 0.2$ kg per cu m.

* Approximate density of saturated steam.

The indicated power of the steam cylinders will be:

$$N_{si} = \frac{1}{75\times 60} k_s \varphi_s \frac{\pi D_s^2}{4} Snp_{si} \text{ metric ihp} \qquad (151)$$

where k_s = number of volumes swept through by all the steam pistons of the pump during one stroke ($k_s=1$ for simplex pumps and $k_s=2$ for duplex pumps).

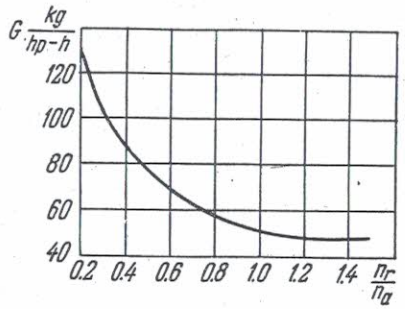


Fig. 74.

The theoretical specific steam consumption per indicated horse power hour is equal to

$$q_{it} = \frac{G_s}{N_{si}} = \frac{60k_s \frac{\pi D_s^2}{4} Sn\,\varphi_s\gamma_s}{\frac{k_s}{75\times 60}\frac{\pi D_s^2}{4} Sn\,\varphi_s p_{si}} = \frac{75\times 60\times 60\gamma_s}{p_{si}} =$$

$$= \frac{27\times 10^4 \gamma_s}{p_{si}} \text{ kg per ihp-h}$$

The actual steam consumption per ihp-h is

$$q_i = \beta_{sc} q_{it} = \beta_{sc} \frac{27\times 10^4 \gamma_s}{p_{si}} \text{ kg per ihp-h} \qquad (152)$$

where β_{sc} = factor taking into account the heat losses and steam condensation in the steam cylinder (Table 23).

Table 23

Type of pump	Indicated power, metric hp					
	1	3	5	15	50	100
Low-speed β_{sc} = ...	2.4	2.2	2.0	1.84	1.75	1.70
High-speed β_{sc} = ...	2.2	2.0	1.8	1.68	1.58	1.55

If the pump operates with back pressure, the steam consumption per hp-h is

$$q_{ib} = \chi_s q_i \qquad (153)$$

where χ_s= 1 for a back pressure of 1 atm (abs)
χ_s= 1.2 for a back pressure of 2 atm (abs)
χ_s= 1.3 for a back pressure of 3 atm (abs).

The steam consumption of direct-acting pumps operating with steam expansion is equal to about 0.65 q_i if the spent steam is exhausted to the atmosphere and to 0.5 q_i if it is exhausted into a condenser.

The steam consumption increases if the load is reduced as can be seen from the curves in Fig. 75 which have been plotted for various values of $\frac{n_r}{n_a}$. The steam consumption of pumps can be reduced by

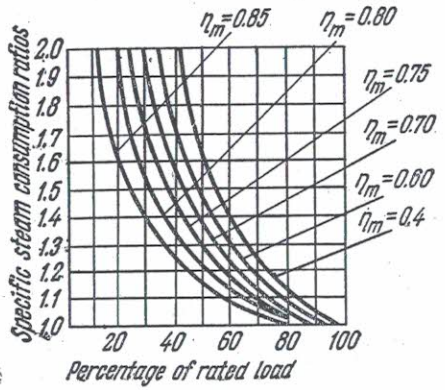


Fig. 75.

increasing the mechanical efficiency, η_m, live steam pressure and/or the number of piston strokes per minute, as well as by reducing the stroke (Fig. 75).

The mean temperature of the steam in the cylinder is raised when the number of piston strokes per minute is increased while a shorter stroke promotes more rapid filling of the cylinder with steam and thereby reduces losses due to steam condensation. The economy of direct-acting pumps can be raised by running them on exhaust steam obtained from the main power plant, or by utilizing the heat in the spent steam from the pumps for domestic needs, or to preheat the feed water.

The advantages of direct-acting and power reciprocating pumps include:

(1) The discharge pressure is limited only by the power of the motive unit and the strength of the pump parts.

(2) The pressure created by the pumps is independent of their capacity.

(3) The pumps can create the specified pressure at any number of piston strokes per minute.

(4) These pumps are capable of operating with dry suction.

Additional features of direct-acting pumps are: (1) simplicity of their construction and maintenance; (2) dependable and quiet operation; (3) simple adjustment of pump delivery by throttling the steam to the cylinder, and (4) the absence of a fire hazard when pumping inflammable liquids.

Drawbacks of reciprocating pumps include:

(1) The inertia forces resulting from the reciprocation of the piston lead to unsteady motion of the liquid in the suction and discharge lines.

(2) The valves, needed for pump operation, increase the cost of the pumps and are the most frequent cause of pump troubles.

(3) Since these pumps run at comparatively low speeds, large units must be built to obtain high capacities.

(4) These pumps are not suitable for handling liquids containing foreign matter.

An additional drawback of direct-acting pumps is their low economy because of high steam consumption (30 to 70 kg per hp-h) when operating at the rated speed, and a considerable increase in consumption at reduced speeds.

Direct-acting pumps operate on saturated or superheated steam at a pressure up to 25 atm (abs) as a rule. They are employed on ships with steam power plants as:

(1) wet-air pumps with capacities up to 60 tons per hour;

(2) circulating pumps with capacities up to 300 tons per hour;

(3) fuel injection pumps with capacities up to 60 tons per hour;

(4) oil, fire, bilge, ballast, and sanitary and drinking water pumps with capacities up to 200 tons per hour;

(5) delivery pumps for various shipboard services with capacities up to 100 tons per hour.

Direct-acting pumps are usually installed to satisfy various shipboard needs on motorships having auxiliary steam boilers.

Single- and double-acting attached pumps are employed on steamships as engine-feed, engine-bilge, wet-air and circulating pumps. On ships with diesel propulsion attached pumps serve the drainage, lubricating and engine cooling systems.

Power reciprocating pumps are most frequently used for pumping water and are found chiefly on motorships.

Double-acting simplex and duplex pumps are used for ballast, drainage, fire and domestic water services.

Single-acting triplex power reciprocating pumps with constant or variable plunger stroke are almost exclusively used for boiler feed.

Hand pumps are employed as auxiliary units for transferring fuel, lubricants and water, as well as for drainage purposes in compartments which are not served by inlet branches of the drainage system.

The models of pumps recommended for use for various purposes are listed in the "Manual of Types and Sizes of Marine Equipment, Machinery and other Articles" issued by the U.S.S.R. Ministry of Sea Transport and in "Compiled Data on Marine Auxiliary Machinery Manufactured in the U.S.S.R." issued by the same ministry.

The main parts of pumps are made of structural steels or ordinary foundry cast irons. Parts that come in contact with sea water are commonly made of bronze. Piston rods, connecting rods, fastening bolts, nuts, keys, and other small parts are made of steel with a carbon content from 0.35 to 0.45 per cent. Parts on which no high requirements are imposed in respect to mechanical strength or tightness may be cast of grey iron grade СЧ12-28 or СЧ 16-36 according to U.S.S.R. Std GOST, 1412-42. Pump parts subject to pressure as, for example, pump housings, cylinders, connections, valve chests and pistons are cast of grey iron grade СЧ 28-48. Critical parts of high-pressure pumps may be cast of steel with a carbon content from 0.3 to 0.4 per cent.

Surfaces subject to wear are made of bronze or are babbitted. The packing of pistons and stuffing boxes may be of cast iron, steel, bronze, plastics, leather, rubber or rubberized fabric depending upon the properties of the liquid being handled.

Valves may be of bronze, stainless steel, leather, rubber or rubberized fabric depending upon the temperature of the liquid being pumped and its chemical and mechanical compositions.

C. ROTARY AND HAND PUMPS

1-18. Rotary Pumps. General Considerations

Rotary pumps are positive displacement pumps in which energy is transmitted from the motive unit to the liquid by an impelling element having only a rotary or a rotary and reciprocating motion in the pump body, or casing.

As the impelling element or elements travel in the casing, certain pockets formed in the latter and communicating with the admission connection, increase in volume while other pockets, communicating with the discharge line, decrease in volume, thus effecting the suction and discharge actions of the pump.

Since the corresponding pockets formed in the casing are constantly connected to the suction and discharge lines, an almost uniform movement of the liquid in these lines can be achieved at constant speed of the impelling elements which are also called rotors. Rotary pumps driven by nonreversible motive units have no valves. Valves are provided when such pumps are powered by the reversible main marine engines to ensure a constant direction of liquid flow in the suction and discharge lines.

The admission cavity of a rotary pump must be separated from the discharge cavity to prevent the liquid from flowing back from the discharge to the suction line and to maintain a discharge head. This is done by:

(1) installing a distributing mechanism;

(2) imparting a suitable shape to the impelling element or internal surface of the casing;

(3) employing two or three impelling elements with pockets in which they carry liquid from the admission to the discharge cavity.

The same impelling elements create conditions necessary for the admission of liquid into the pump from the suction line, and for delivery to its destination along the discharge line.

Rotary pumps are finding ever wider applications in shipboard service for pumping liquids with a higher viscosity than that of water. This is due to the following main advantages of these pumps:

(1) small overall size;

(2) comparatively low weight;

(3) comparatively high uniformity of liquid discharge;

(4) high rotor speeds (up to 5,000-10,000 rpm) that enable them to be linked directly to high-speed motive units of any type;

(5) capability of creating a high head (up to 300 and in some cases even to 750 kg per sq cm);

(6) capability of operating with dry suction by creating a vacuum of from 500 to 640 mmHg;

(7) capability of operating in various positions;

(8) comparatively high mechanical and volumetric efficiencies.

Rotary pumps are mainly employed:

(1) for pumping liquid fuels;

(2) as booster pumps in fuel transfer systems;

(3) as drainage pumps of fuel tanks;

(4) as oil transfer pumps and pumps for oil circulation systems;

(5) as circulating pumps of internal-combustion engines;

(6) as vacuum pumps;

(7) as general shipboard service pumps handling, not only high-viscosity liquids, but those of lower viscosity as well (water, kerosene and gasoline);

(8) as pumps supplying hydraulic steering gear.

In their design rotary pumps are classified as:

(1) lobe-type;

(2) eccentric;
(3) vane (with guided blades or vanes);
(4) gear-type;
(5) helical screw;
(6) water-ring;
(7) rotary piston.*

In respect to the number of rotors, rotary pumps may be single-, double- or triple-rotor types. Pumps are also classified as reversing or nonreversing rotary pumps depending upon the direction of rotor rotation.

1-19. Typical Constructions and Capacities of Rotary and Hand Pumps

(A) LOBE-TYPE ROTARY PUMPS

The lobe-type pump has two rotors, each with either two or three lobes. Three-lobe pumps usually have a higher volumetric efficiency than the two-lobe type but are difficult to manufacture.

The principle of a two-lobe pump is shown schematically in Fig. 76. The rotors *4* are arranged in the oval casing *1*. The rotors are driven by a motive unit and their rotation is coordinated by toothed gearing arranged outside of the casing.

At the moment shown in Fig. 76, liquid is being drawn into pocket *5*, formed between the casing and the rotors, from the admission connection *2* and is being discharged from pocket *6* through the discharge connection *3*. In pocket *7* suction has been completed. Upon further rotation of the rotors the liquid in pocket *7* will be forced into the discharge line while suction will be completed in pocket *5*.

An essential disadvantage of lobe-type pumps is that the rotors must be precisely fitted to each other and to the working surface of the casing. Slightest errors in fitting the working surfaces or inaccuracy in assembly lead to premature wear of the impelling elements, and a reduction in the volumetric efficiency and in the head created by the pump. The working surfaces of the rotors as well as those of the rotors and casing must not be in actual contact, i.e., a certain definite clearance must be maintained.

This clearance prevents premature wear of the surfaces but lowers the volumetric efficiency because it increases the leakage of liquid from the discharge to the admission cavities. An increase in clearance reduces the head produced by the pump.

In one revolution of the rotors, four working volumes, i.e., the volumes of four pockets, are delivered into the discharge line. Therefore, if $F_p \approx 0.2\,D_r^2$ sq m and the casing has a length of l_c m, while the volumetric efficiency is $\eta_v = 0.70$ to 0.80 and the rotor diameter is D_r m, then at a speed of n rpm the pump capacity, with sufficient accuracy for practical purposes, will be

$$Q = 60 \times 4F_p l_c \eta_v n = 60 \times 4 \times 0.2 D_r^2 l_c \eta_v n = 48 \eta_v D_r^2 l_c n \text{ cu m per hour}$$

The length l_c can be determined by the ratio

$$\varphi_c = \frac{l_c}{D_r} \approx 0.6 \text{ to } 1.5$$

An increase in the factor φ_c reduces the volumetric efficiency due to increased leakage, or slip, while the difficulties in machining the parts and the overall size of the pump will increase.

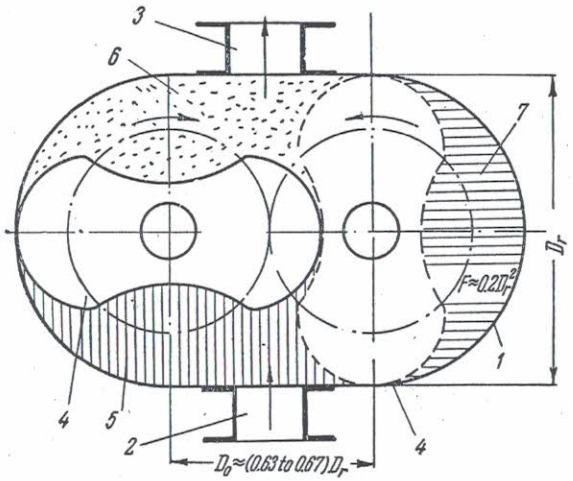


Fig. 76.

(B) ECCENTRIC PUMPS

The impelling elements of eccentric pumps are the eccentrics, yokes and rings which are driven with a rotary and reciprocating motion by the pump shaft on which they are either freely mounted or keyed.

These pumps may be of the single-shaft type with one or several impelling elements. The greater the number of elements, the higher the capacity will be and the more uniform the delivery.

A general view of a single-shaft triple-eccentric shoe-type pump and its principle of operation are illustrated in Fig. 77*a*, *b*, *c*, *d* and *e*.

Upon rotation of the shaft *1* with its rigidly mounted eccentrics *2*, shoes *3*, mounted freely with their slots embracing the faces of the

* Rotary piston pumps are used in hydraulic steering gear and will be discussed in connection with them.

eccentrics, reciprocate vertically. The faces *4* of shoes *3* slide along the seat *5* of cover *6* which has three ports *7*, *8* and *9* for each shoe. In the shoes are the passages *10* and *11* which communicate with the inner slots. The seat *5* of cover *6*, the internal surfaces of the shoe *3* and the surfaces of the eccentric comprise a separate working volume, or pocket, *15*.

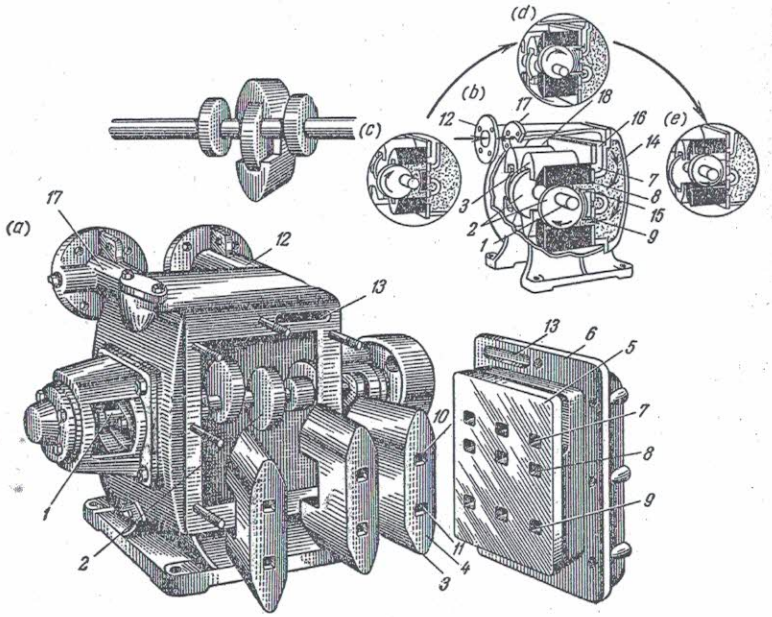


Fig. 77.

As the eccentric rotates in the clockwise direction, in the position shown in Fig. 77*b*, liquid is drawn into pocket *15* through connection *12*, port *13*, cavity *14* of cover *6*, port *8* and passage *10*, and port *9* and passage *11*.

Upon further clockwise rotation of the eccentric, it raises shoe *3* whose face closes the ports *8* and *9* (Fig. 77*c*).

As soon as these ports are closed the suction stroke is finished and the discharge stroke of the eccentric begins (Fig. 77 *d*). Now the liquid passes from the pocket *15* through passage *10* of the shoe and the discharge port *7* of the cover into cavity *16* which communicates with cavity *18* of the pump casing and, through it, with the discharge connection *17*. Still further rotation of the eccentric lowers the shoe so that its face closes port *7* and uncovers ports *8* and *9* (Fig. 77*e*),

thus beginning the next suction stroke. The shoes are held tightly against seat *5* by the difference in pressure in the suction cavity of cover *6* and cavity *18* in the pump casing.

In a triple-eccentric pump the eccentricity of each unit is arranged at an angle of 120° with those of the other eccentrics. This ensures highly uniform delivery of the liquid.

More uniform delivery and a higher capacity is achieved by arranging the eccentrics in pairs or in groups at angles of 180° to each other, and the pairs or groups at angles of 120° to the other pairs or groups. This arrangement also reduces the loads on the bearings and the bending moment on the shaft.

The bearing liners of pumps handling water are made of rubber which is lubricated by the water being pumped. The shoes are made of a plastic faced with synthetic rubber on the sliding surface.

Such pumps may be employed to handle aggressive or highly viscous liquids at any temperature, if the proper materials are selected for the pump parts.

The capacity of shoe-type pumps ranges from 1.3 to 80 cu m per hour at a speed of the eccentric up to 1,250 rpm. They can operate against a head up to 75 mH_2O and have a suction lift from 5.5 to 8 mH_2O depending upon the working conditions and the viscosity of the liquid.

The theoretical displacement V_t of any eccentrically mounted impelling element in one revolution of the shaft is equal to the volume described in rotation about the shaft axis of a plane surface with a length equal to the radius R_e of the impelling element plus the eccentricity e, and a width l_e equal to the thickness of the impelling element, minus the volume of the impelling element itself. Thus

$$V_t = \pi (R_e + e)^2 l_e - \pi R_e^2 l_e = \pi [(R_e + e)^2 - R_e^2] l_e \text{ cu m}$$

At a shaft speed of n rpm, volumetric efficiency $\eta_v = 0.75$ to 0.85 and with z impelling elements, the hourly capacity of the pump will be

$$Q = 60 V_t \eta_v z n = 60\pi [(R_e + e)^2 - R_e^2] z l_e \eta_v n \text{ cu m per hour}$$

(C) VANE PUMPS

In vane pumps the impelling elements are vanes which slide freely in slots of the rotor. The rotor rotates in the casing in which it is arranged eccentrically in respect to the internal surface.

In some vane pumps, in which the rotor is concentrically arranged in the casing, the inner surface of the latter is of elliptical shape.

Vane pumps may be either single or double action and have from 2 to 12 vanes.

The pump illustrated in Fig. 78*a* and *b* has a rotor *2*, eccentrically arranged in the casing *13*, with slots in which vanes *7* slide. The vanes have pins *11* on which shoes *10* are fitted.

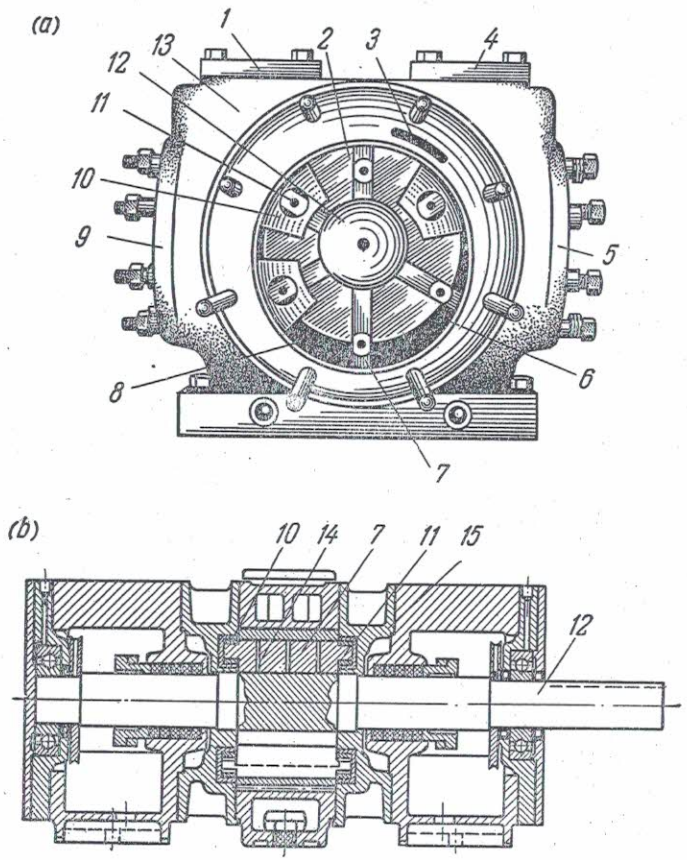


Fig. 78.

Rotor *2* and the vanes *7* rotate together with the shaft *12* while the shoes of the vanes slide in slots of the covers *15* and move the vanes in a radial direction. If the rotor rotates in the counterclockwise direction, liquid is admitted through connection *9* into pocket *8*, and is discharged from pocket *6* through connection *5*.

After the space between the vanes passes the discharge port, the liquid in this space would be compressed if the space were not connected to another port *3* communicating with the discharge cavity. This precludes compression of the liquid and increases the displacement of the pump. A relief valve is connected to flanges *1* and *4*. If the pressure is too high in the discharge cavity, the valve opens and bypasses liquid to the suction cavity.

Grooves *14* are provided on the vanes to prevent compression of the liquid in the slots as the vanes slide into them. These grooves also increase the displacement of the pump since through them the slots fill with liquid when the vanes slide outwards during suction and then this liquid is forced out into the discharge cavity during discharge. Thus the vanes operate similar to the pistons of reciprocating pumps.

The pressure created by such pumps reaches 7 atm (gauge) but may be even higher (20 atm) if the clearances between the end surfaces of the vanes and the inner surface of the casing are reduced.

As a rule pump shaft speeds range from 20 to 300 rpm with capacities from 2.7 to 5.7 cu m per hour, but higher speeds and capacities are possible.

The displacement of the pump during one revolution of the rotor is equal to the volume of the working chamber, or pocket, minus the volume of the vanes in the chamber. If it is assumed that the volume of liquid delivered by the vanes, due to their radial travel in the rotor slots, is approximately equal to the volume of the working section of the vanes, then the effect of the latter on the reduction in displacement can be neglected.

At a rotor diameter D_r, casing bore diameter D_c, rotor length l_r, rotor speed of n rpm and a volumetric efficiency η_v, the pump capacity is

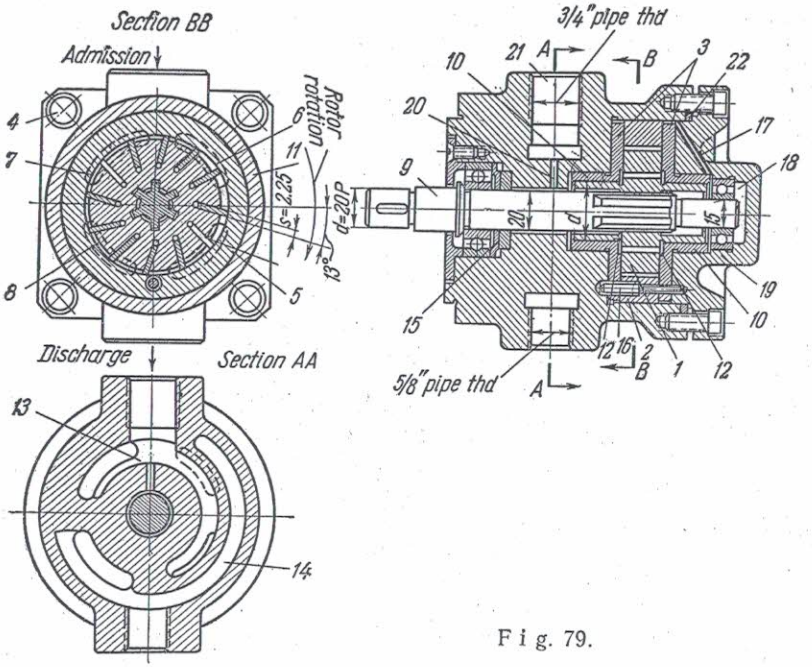
$$Q = 60\left(\frac{\pi D_c^2}{4} - \frac{\pi D_r^2}{4}\right) l_r n \eta_v = \frac{60\pi}{4}(D_c^2 - D_r^2)\, l_r n \eta_v \text{ cu m per hour} \quad (154)$$

Double-action vane pumps with a large number of vanes are used to obtain higher capacities with smaller overall size and a more uniform discharge.

A double-action vane pump is shown in Fig. 79. These pumps can operate against a pressure up to 70 kg per sq cm. Their small size, dependability in operation and simple design have found them wide application in installations requiring uniform liquid flow at a high head.

The pump casing *11* has fastening holes *4* for mounting the pump in the ship and contains the cam ring *2* with an internal contour of special elliptical shape. Rotor *1* is mounted on the splined drive

shaft *9* with considerable clearance and has slots for the vanes at an angle of 13° to the radius of the rotor. Shaft *9* runs in ball bearings while the body of the rotor runs in bushings *10* of the bronze valve plates *3*, which form the side surfaces of the pumping chamber. The loose fit of the rotor on the shaft permits its proper alignment in its



F i g. 79.

bearings and simplifies pump assembly. The stepped ring pin *16* fixes the cam ring *2* and valve plates *3* in respect to the pump casing.

By means of the ports *6* and *8* in the valve plates *3*, the suction chamber is connected to passage *13* while the discharge chamber is connected by means of similar ports *5* and *7* to passage *14*.

Upon rotor rotation, the vanes in the slots are stationary as they pass the seal between ports *5* and *8*. As it approaches the edge of port *8*, each vane begins to slide outwards due to the shape of the cam ring, the centrifugal force acting of the vane, and pressure of the liquid delivered from the discharge chamber under the vanes through grooves *12* in the bronze valve plates *3*.

Thus held against the inner surface of the cam ring whose contour they follow, adjacent vanes form a pocket, in conjunction with other parts of the pump, with a gradually increasing volume. This creates

a vacuum and the pocket is filled with liquid admitted through port *8*. Suction continues until the adjacent vanes reach a position in which their surfaces enclose a volume extending between the edges of ports *8* and *7* and bounded by the cam ring and rotor. Discharge through port *7* begins upon further rotation of the rotor. The volume of the given pocket begins to decrease from this point as the vanes are pushed into their slots due to the contour of the cam ring. Discharge ends when the adjacent vanes reach a position between the edges of ports *7* and *6*. Still further rotation of the rotor results in suction through port *6* and discharge through port *5*. Thus, in one revolution of the rotor each pocket formed by two adjacent vanes will discharge twice. For this reason, this pump is called a double-action pump.

The arrangement of the discharge (and admission) ports directly opposing each other balances the pressure of the liquid on the rotor. Consequently, the rotor bearings are subject only to a small load and may be of light design and minimum size. The pressure exerted by the vanes on the cam ring, especially as they pass through the sections where the admission ports are located, is reduced by making the vanes quite thin (about 2.25 to 2.5 mm).

Liquid, leaking through the joint between the rotor and the valve plates *3*, flows through passage *17* into space *18* and then along the shaft and passage *20* into the admission connection *21*. Cork ring *15* and cork gasket *22* are used as packing on the pump shaft and between the head *19* and casing *11*.

The rotor and cam ring of such pumps are made of cyanided alloy steel and the vanes of high-speed steel. This is dictated by the operating conditions and considerations of wear resistance. No high quality materials are required for the other parts of the pump.

The theoretical capacity of a double-action vane pump can be calculated with sufficient accuracy by the formula:

$$Q_t = 2l_r n \left[\pi (r_2^2 - r_1^2) - \frac{(r_2 - r_1)\,\delta z}{\cos \alpha^\circ}\right] 60 \text{ cu m per hour} \quad (155)$$

where l_r = rotor length, m
n = rotor speed, rpm
r_2 = semimajor axis of the cam ring contour, m
r_1 = semiminor axis of the cam ring contour, m
δ = thickness of the vanes, m
α° = angle vanes are inclined to the radius, deg
z = number of vanes.

Rotor shaft speed, n, of these pumps may reach 3,000 rpm, the vane inclination, α°, ranges from 6° to 13°. The velocity of the liquid drawn through the casing ports varies from 1.5 to 2 m per sec, the volumetric efficiency, η_v, from about 0.82 to 0.98 and the mechanical efficiency, η_m, from 0.6 to 0.92.

(D) GEAR PUMPS

Gear pumps are employed to transfer liquids both of high and low viscosity. Mainly gear pumps are used in the lubrication circulation systems of various shipboard installations. The reasons for their wide application for this purpose are their fairly uniform delivery, simple manufacture, dependable service, comparatively high capacity for their small overall size, and their convenient installation in comparison with other types of pumps.

Gear pumps consist of a driver and a driven gear enclosed in a casing and in mesh with each other. Both gears may be external or one of them may be an internal gear. The gears may be of either the spur, helical, herringbone or spiral type. The tooth profile is usually of the involute type, such gears being more convenient in manufacture and in making corrections to avoid interference in meshing. The use of one internal gear in the pump will reduce its size to some extent. The overall size of gear pumps increases with the number of teeth on the gears. In addition to a reduction in pump size, gears with less teeth also have a tendency to reduce the compression of the liquid in the tooth spaces at the point where the gears mesh. On the other hand, gears with a smaller number of teeth mesh under more adverse conditions, the teeth have lower strength and, in general, pump performance is poorer.

The operation of a gear pump is based on the transfer of the liquid in the tooth spaces from the admission to the discharge sides along the walls of the casing, upon rotation of the gears. The teeth of gears that are in mesh separate the admission and discharge chambers of the pump from each other. As they come out of mesh in the admission chamber, the teeth free the tooth spaces into which the liquid is admitted. The teeth going into mesh in the discharge chamber force the liquid out of the tooth spaces and serve, at the same time, as a seal preventing liquid flow from the discharge back to the admission chamber between the gears. The very small clearance between the teeth and the internal surfaces of the casing (in the order of several hundredths of a millimetre) almost completely excludes liquid slip back to the admission chamber along the casing. The high pressure in the discharge chamber is maintained by the continuous delivery of liquid by the teeth entering this chamber.

The construction of a single-stage two-gear pump is illustrated in Fig. 80. The casing *4* of the pump encloses the driving gears *3* and *5* and the driven gears *21* and *20*. Gears *3* and *20* have right-hand helical teeth while gears *5* and *21* have left-hand helical teeth. Gears *3* and *5* are keyed to the drive shaft *12* and form a single herringbone gear, as do gears *21* and *20*, mounted on driven shaft *17* and meshing with the first two gears. Gear *21* is keyed to shaft *17*

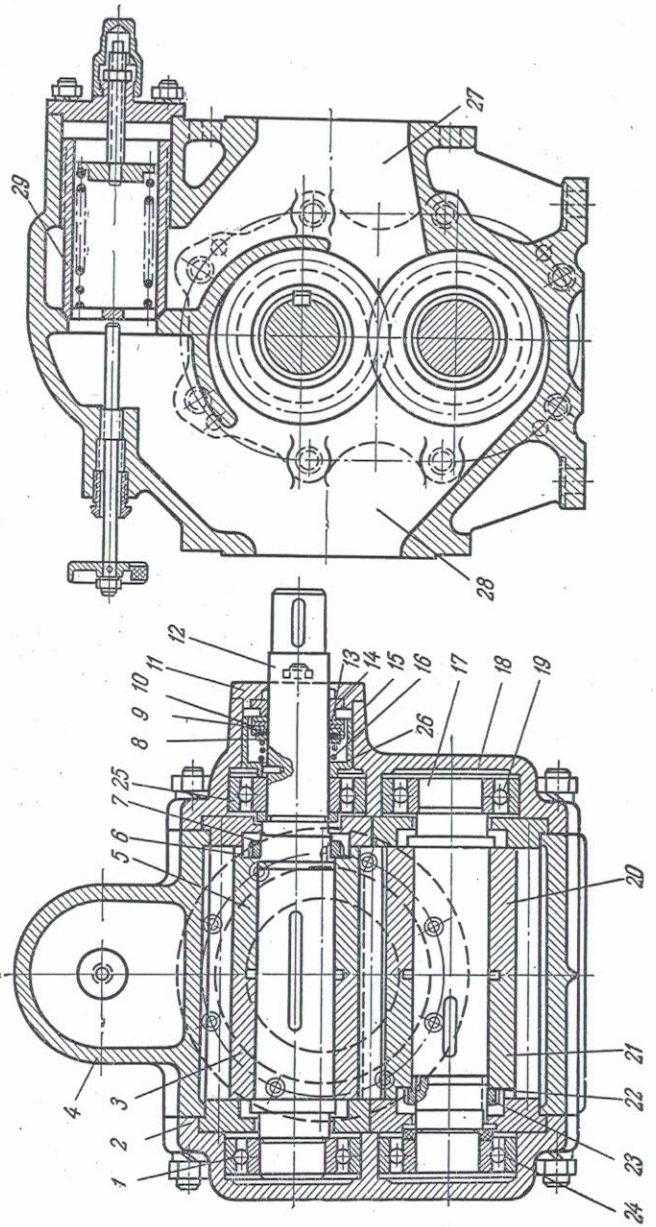


Fig. 80

while gear *20* is freely mounted, thereby enabling the latter to align itself in respect to the teeth of gear *5*. Ball bearings *1*, *19*, *24* and *25* are press-fitted on the journals of the rotors. The gears are held on the shafts by round nuts *7* and *23* which are locked by the lugs of the lock washers *6* and *22*. The bearings are fitted into the recesses of the covers *18* and *2* which form the side walls of the casing. A packing gland seals the drive shaft where it extends from the casing. It consists of the shell *15* into which are inserted sleeve *26* fitted on the shaft *12*, steel thrust ring *8*, rubber sealing ring *9*, bronze ring *10* and steel step bearing *14* with the rubber buffer ring *13*. When cover *11* of the gland is fastened in place, spring *16*, inserted between sleeve *26* and thrust ring *8*, compresses rubber ring *13*, sealing the surface of shaft *12*.

The driving gears rotate in the counterclockwise direction, delivering liquid from chamber *27* to chamber *28*. Relief valve *29* (which is also a bypass valve) prevents the pressure in chamber *28* from rising above the permissible value.

Attached gear pumps of reversible marine engines are equipped with special valves which maintain liquid flow in a single direction without regard for the direction of drive shaft rotation.

An attached gear pump of the main engine oil system is shown in Fig. 81.

Here, casing *1*, divided into two compartments by a partition, encloses the meshing gears *2*, *3*, *4* and *5* which actually constitute two separate pumps housed in a single casing.

The left half of the pump with gears *2* and *3* transfers oil from the engine crankcase through the filter and cooler into the service tank. The right half of the pump with gears *4* and *5* draws oil from the service tank and discharges it into the oil system of the engine. Driving gear *2* is integral with shaft *37* while gear *4* is keyed on this shaft.

Driven gear *3* is also integral with its journals and shaft while gear *5* is freely mounted on the right end of this shaft.

Each half of the pump has its ball-type inlet valves *21* and *23*, and discharge valves *20* and *22*, as well as separate inlet chambers *24* or *25* and discharge chambers *12* or *15*.

The halves of the pump are identical in all respects. Upon clockwise rotation of shaft, in the left half of the pump the liquid admitted to the inlet chamber *25* passes through inlet valve *21* into chamber *36* and is discharged into chamber *15* through discharge valve *22*.

If drive shaft rotation is reversed, admission will be through valve *23* and discharge through valve *20*. Liquid is delivered from the pump through passage *26* provided in the cover *33*, and connection *27*.

Pressures higher than permissible in the discharge chamber are prevented by the relief valve *30* which, when open, connects chamber

15 to chamber *25* through passage *31* provided in cover *34*. The liquid is admitted into the right half of the pump through connection *28*.

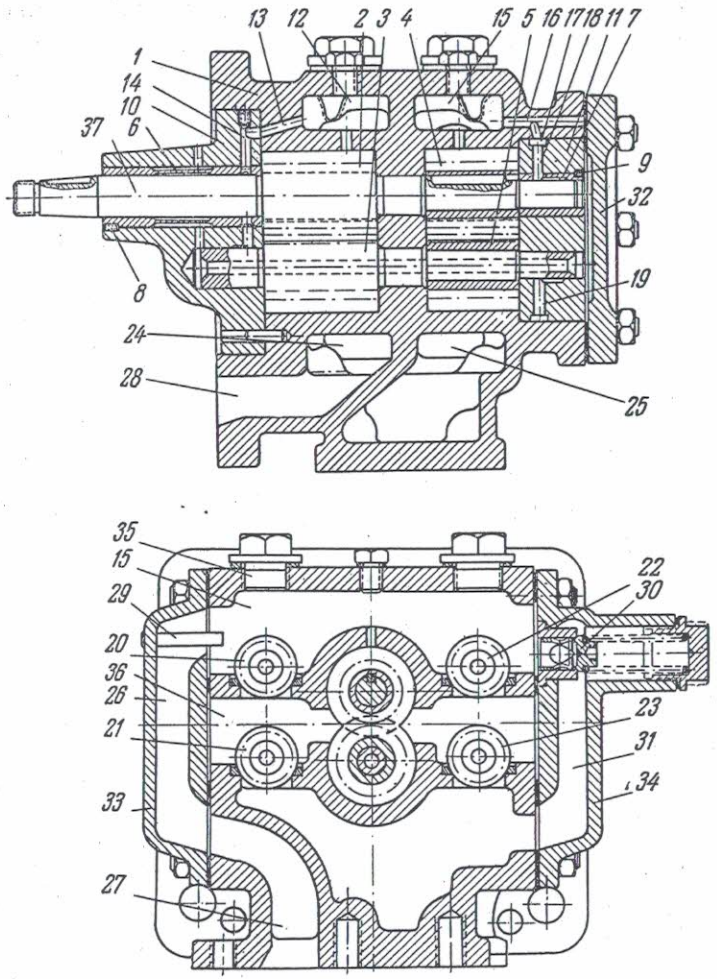


Fig. 81.

Bronze bushings *6* and *7* are fitted into covers *10* and *11* where they are secured by set screws *9* and *8*. These bushings serve as the bearings of the drive shaft. Lubricating oil is supplied to the bearings from the discharge chambers along the passages *13*, *14*, *16*, *18* and *19*.

Oil is delivered to passages *18* and *19* through the annular groove *17*. On the right side the pump is closed by cover *32*.

The holes closed by threaded plugs *35* provide access to lap the seats of the valves. Retaining pins *29* prevent the valve balls from being carried away into the discharge line.

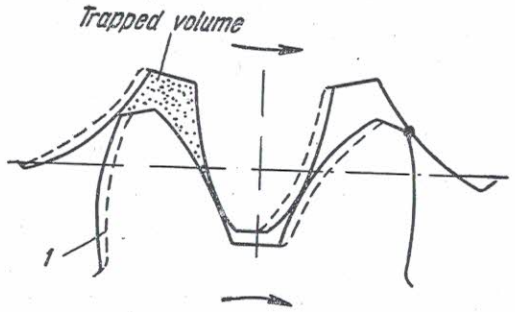


Fig. 82.

When the gear teeth are made with a precise fit they mesh so tightly with one another that a part of the liquid is locked in the tooth space or must escape through a very small clearance between

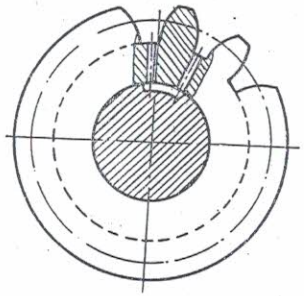


Fig. 83.

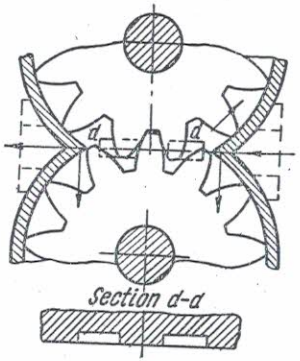


Fig. 84.

the teeth in engagement (Fig. 82). In this manner a high pressure is developed in the trapped liquid which impedes gear rotation and leads to a loss in power, additional load on the bearings, spreading of the gears and heating of the liquid.

This defect can be eliminated in pumps with a small number of teeth by providing a clearance, or backlash, of from 0.2 to 0.5 mm normal to the tooth profiles and by making corrections in the tooth

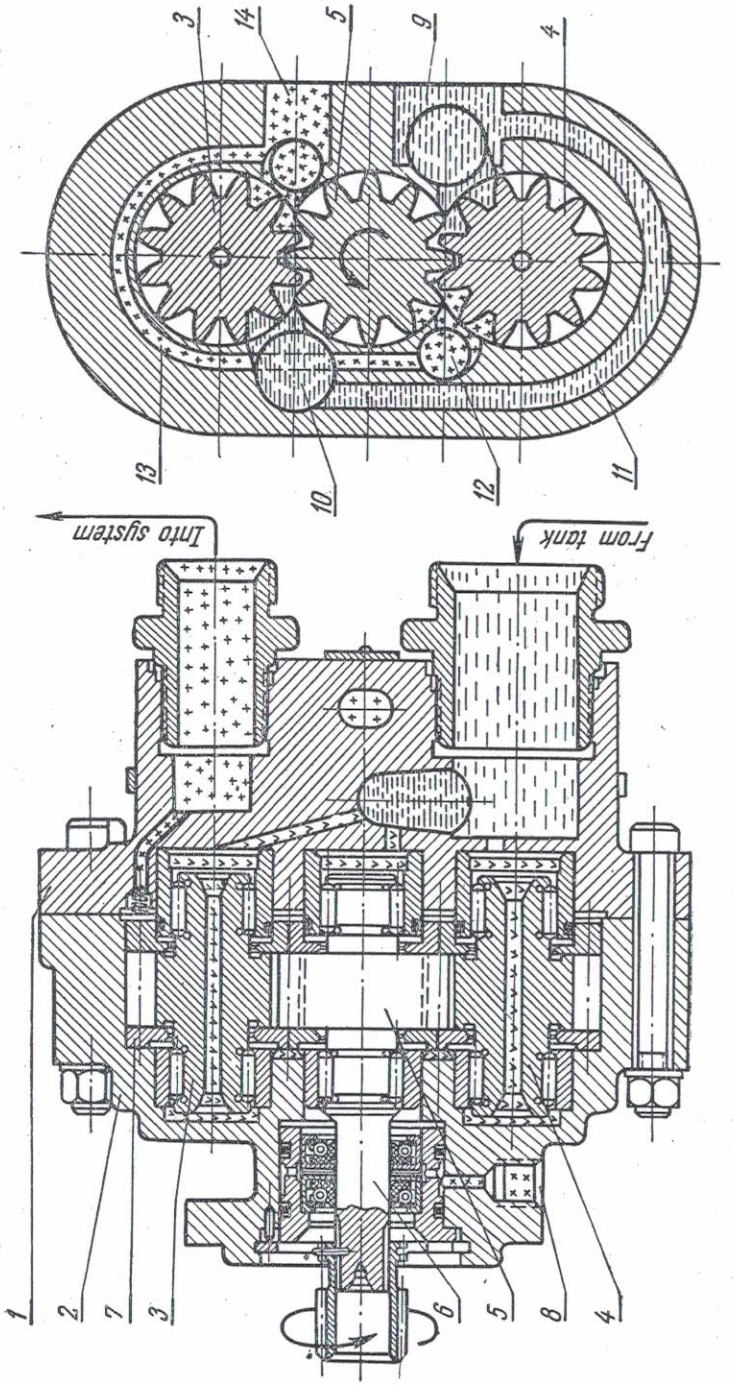


Fig. 85.

profile. The same purpose is served by making grooves *1* (Fig. 82) along the tooth profile, by connecting the tooth spaces that are in engagement through grooves with the discharge chamber (Fig. 83), or by providing connecting recesses in the end covers of the pump (Fig. 84). Other methods are also employed to prevent compression of the liquid in gear pumps.

Parallel engagement of the gears is used to increase the pump capacity without greatly increasing the overall size.

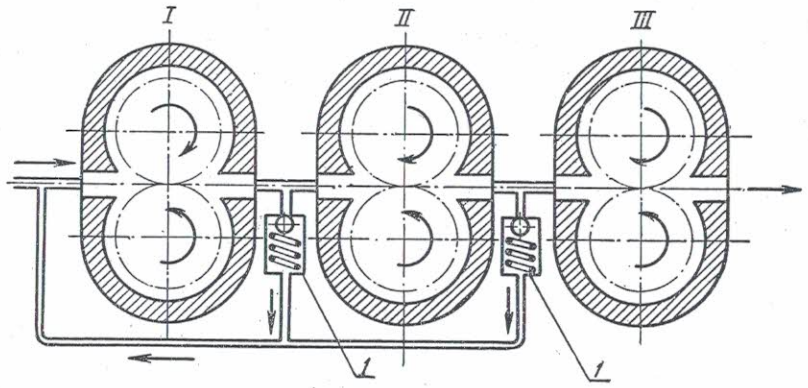


Fig. 86.

The principle of a single-stage double-flow triple-gear pump is shown in Fig. 85. The pump casing comprises the two covers *1* and *2* held together by bolts and nuts. It encloses two driven gears *3* and *4* and the driving gear *5*. The driven gears are made integral with their journals which run in needle bearings. The driving gear is integral with its shaft *6*. The faces of the gears are sealed by floating bushings *7* which are held against the gears by the pressure of the liquid. Shaft *6* is provided with a liquid seal, the liquid being delivered from the discharge chamber to the inlet connection *8*. When the middle gear *5* rotates in the counterclockwise direction both driven gears rotate clockwise. Liquid is admitted to gear *4* through connection *9* and to gear *3* from chamber *10*, which is connected by passage *11* to the inlet connection *9*. Liquid is discharged from gear *3* to connection *14* and from gear *4* to chamber *12* and further through the passage *13* to the discharge connection *14*.

The capacity of a triple-gear pump is twice that of a double-gear pump having the same size of gears.

The pressure created by gear pumps can be increased by the use of multiple-stage pumps in which a single casing houses two or more double-gear pumps connected in tandem.

The principle of a three-stage pump with stages *I*, *II* and *III* and a relief valve *1* is shown schematically in Fig. 86. Its construction is shown in Fig. 87.

The capacity of each stage is higher than that of the next stage and, therefore, each stage has a relief valve set to a definite pressure to bypass the surplus liquid.

The pump casing comprises the two covers *1* and *2* and two cylindrical rings *3* and *4* between which annular partitions *5* and *6* are inserted. The parts of the casing are held together by bolts and nuts

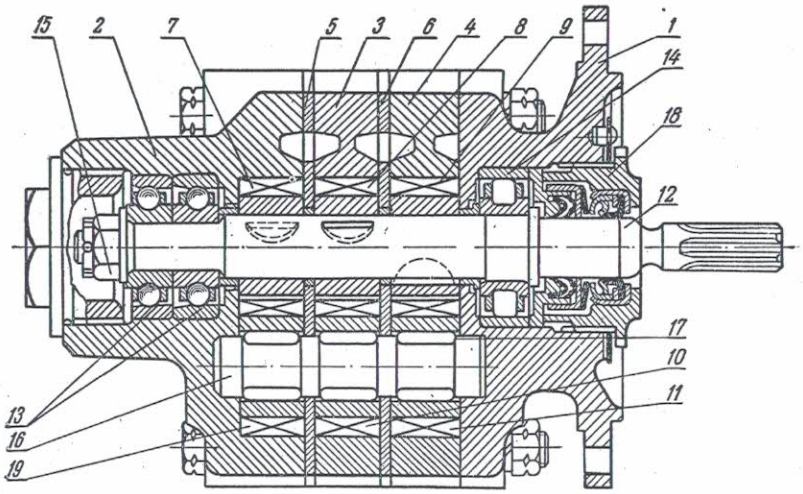


Fig. 87.

and form three chambers for the driving gears *7*, *8* and *9*, and the driven gears *19*, *10* and *11*. The driving gears are secured on the shaft *12* with Woodruff keys. The shaft runs in ball *13* and roller *14* bearings which are secured on the shaft by the nut *15*. The driven gears run on needle bearings *17* which are mounted on shaft *16*. The discharge chamber of each stage serves as the admission chamber of the next stage. Where the shaft *12* extends from the casing it is sealed by a double press-fit seal consisting of packing cups and shells in the thrust bushing *18*.

In another type, the internal gear pump, the driven gear is arranged inside the driving gear. This substantially reduces the overall size of the pump and leads to better filling of the tooth spaces with liquid.

Fig. 88 illustrates an internal gear pump in which the driving gear rotates in the counterclockwise direction. The pump casing,

consisting of the covers *6* and *7* and ring *8* fastened together by screws *9*, houses the internal gear, or rotor, *5* made integral with its shaft *10*. Sealing shoe *1* is held tightly from above against the outer cylindrical surface of gear *5* by the spring *11*. A similar shoe *2* is held from below in the same manner by spring *12*. A crescent seal *3* is fitted to the inner surface of the driving gear *5* and to the outer surface of the teeth on driven gear *4*. The three seals *1*, *2* and *3* separate the admission *14* and discharge *15* chambers from each other. As in external gear

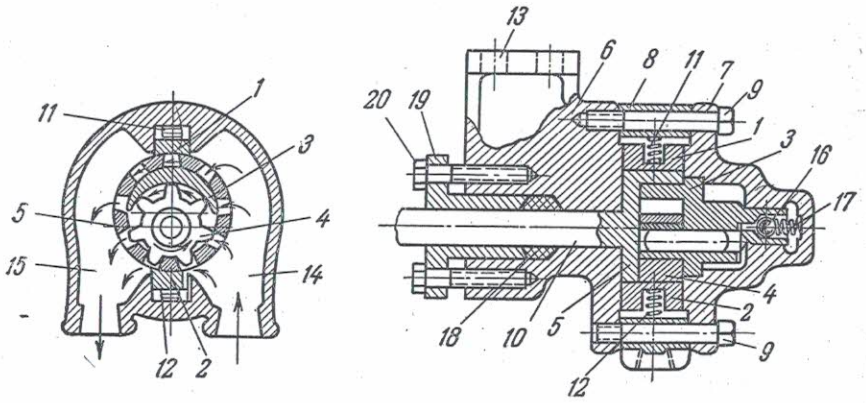


Fig. 88.

pumps, the liquid is carried over in the tooth spaces. A ball-type relief valve *16* loaded by spring *17* is provided to prevent the pressure in the discharge chamber from rising above the permissible value. When the relief valve is open the discharge chamber *15* communicates with the admission chamber *14*. Shaft *10* is sealed in cover *6* by soft packing *18* which is compressed through gland *19* by screws *20*. Flange *13* serves to mount the pump in the ship.

The capacity of a gear pump depends to a great extent upon how completely the tooth spaces are filled with liquid as they pass through the admission chamber in the pump casing. This filling of the tooth spaces, in turn, depends upon the following:

(1) the speed of rotation of the gears, i. e., the time it takes for the tooth space to pass through the admission chamber;

(2) the design of the passages through which the liquid flows to the admission chamber;

(3) the velocity of the liquid in the admission passages;

(4) the pressure in the admission chamber;

(5) the viscosity of the liquid being pumped;

(6) the presence of air and/ or gases in the liquid.

Incomplete filling of the tooth spaces:

(1) lowers the volumetric efficiency and, therefore, the capacity;

(2) emulsifies the liquid;

(3) impairs pump performance and increases wear of the pump parts.

All of these detrimental effects are due to the fact that the tooth space, when not completely filled in the admission chamber, will fill with liquid when it reaches the discharge chamber.

Thus, a reverse flow of liquid to the tooth spaces is formed. This results in a reduction in pump capacity, pulsating pressure in the discharge line, and a pulsating load on the gears and their bearings, leading to premature wear. A pulsating pressure may cause vibration of the components of the pipeline, installations and instruments of the system and impair their normal operation or even lead to leakage in the joints in various parts of the system. Complete filling of the tooth spaces can be ensured if the peripheral speed at the pitch circles of the gears does not exceed 6 m per sec with a pressure in the admission chamber not below 5 mH_2O. In addition, the liquid must be admitted into the tooth space along its whole length and the liquid velocity in the admission passages must not exceed 2 or 3 m per sec. Since gear pumps do not have a high suction lift, it is best to operate with negative lift.

At high gear speeds the liquid filling the tooth space acquires the same speed as the gear and, due to the centrifugal force, it has a tendency to fly out of the tooth space. If the pressure developed by the centrifugal force is higher than that in the admission chamber, the liquid will not enter the tooth spaces.

The theoretical displacement of a double-gear pump in one revolution of the gears should be equal to the volume of all the tooth spaces of the meshing gears. If we assume that the volume of the tooth spaces is equal to the volume of the teeth, then in one revolution of the two gears the pump will deliver a volume of liquid V_g, equal to that described by the working surface of a tooth. Thus

$$V_g = \pi D_{dc}(D_{ac} - D_{dc})\,b \text{ cu m}$$

where D_{dc} = diameter of the dedendum circle of the gears, m

D_{ac} = diameter of the addendum circle of the gears, m.

If the pump gears run at a speed of n rpm, the tooth length is b and the volumetric efficiency $\eta_v \approx 0.7$ to 0.9, the actual hourly capacity will be

$$Q = 60\eta_v V_g n = 60\eta_v \pi D_{dc}(D_{ac} - D_{dc})\,bn \text{ cu m per hour} \quad (156)$$

The small disagreement of the results calculated by this formula with those obtained experimentally enables it to be applied in practice to solve various problems, as well as in pump selection.

In calculating the capacity of gear pumps when the number of teeth on each gear $z=6$ to 12, it is necessary to take into consideration that the volume of the tooth space is slightly greater than that of the tooth. This is done by substituting a factor of 3.5 for π; then

$$Q=60\times 3.5\eta_v (D_{ac}-D_{dc}) D_{dc} bn=$$
$$=210\eta_v D_{dc} (D_{ac}-D_{dc}) bn \text{ cu m per hour}$$

The power required to drive gear pumps is

$$N=\frac{10{,}000Qp_d}{\varepsilon 3{,}600\times 75\eta_m}=\frac{Qp_d}{27\varepsilon\eta_m} \text{ metric hp} \qquad (157)$$

where p_d = discharge pressure, kg per sq cm
η_m = 0.95 to 0.98=mechanical efficiency of the pump
ε = reserve power factor.

The reserve power factor ε, taking into account the necessary extra power in accordance with the viscosity of the liquid, can be found from the following formula:

$$\varepsilon=\frac{1}{1+\frac{^\circ E v_g}{24.2 p_d}} \qquad (158)$$

where $^\circ E$ = viscosity of the liquid in degrees Engler at the pumping temperature
v_g = peripheral speed at the pitch circle of the gears, m per sec
p_d = discharge pressure, kg per sq cm.

The capacity of a gear pump can be increased only by increasing the speed or the size of the gears. The overall size of the pump can be reduced without affecting its capacity by reducing the number of teeth on the gears. Recommended values range from 6 to 10 teeth.

The pressure created by gear pumps does not usually exceed 25 atm, but recently special pumps have been designed with narrower manufacturing tolerances and with parts made of high-quality steels which have reached pressure ratings up to 110 atm. It becomes inexpedient to operate gear pumps at higher pressures due to the high leakage through the clearances.

Tentative dimensions of the pump gears can be calculated from the following approximate formulas.

The pitch diameter of the driving gear can be determined from one of the two following formulas:

$$P.D.=12.3\sqrt[3]{\frac{Q_{lm}}{n}} \text{ mm}$$

$$P.D.=0.93\sqrt[3]{\frac{Q_{lm}}{v_g}} \text{ mm}$$

where Q_{lm} = actual capacity of the pump, litres per min
v_g = peripheral speed at the pitch circle, m per sec.

The module of the gears* $m=0.6+0.1\ P.D.$

The number of teeth on the gears $z_1=z_2=\frac{P.D.}{m}$.

The face width of the gears (tooth length) $b\approx P.D.$ to $(P.D.+8$ mm).

The recommended number of teeth $z=6$ to 10 leads to interference and undercutting of the dedendum of the teeth which impairs gear engagement and, consequently, pump operation.

This highly unfavourable feature is avoided by correcting the gear. This eliminates undercutting and, at the same time, it improves pump operation because it reduces the angle of action, thereby almost completely excluding the trapping of the liquid in the meshing tooth spaces and heating of the liquid.

Moreover, a pump with corrected gear teeth is more wear resistant and has a higher mechanical efficiency due to the reduction in the contact pressure between the sliding surfaces of the meshing teeth and in the pressure of the compressed liquid trapped in the tooth spaces. The profiles of corrected teeth have less curvature and, therefore, fit more tightly together than do normal teeth. This reduces the slip of liquid between meshing teeth and thus increases the volumetric efficiency.

The following formulas can be used to determine the principal dimensions of gears with corrected tooth profile. The diameter D_{ac} of the addendum circle and D_{dc} of the dedendum circle are

$$D_{ac}=P.D.+2(m+k_c m)=P.D.+2m(1+k_c)$$

and

$$D_{dc}=P.D.-2m(1.2-k_c)$$

where m = module of the gears, mm
k_c = correction factor which is taken according to the number of teeth from Table 24.

Table 24

z	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
k_c	0.48	0.46	0.44	0.42	0.40	0.38	0.36	0.34	0.32	0.30	0.28	0.26	0.24	0.22	0.20

The addendum h_a and the dedendum h_d are

$$h_a=m(1+k_c)$$

* According to data compiled by E. Yudin m=(0.24 to 0.44) $\sqrt{Q_{lm}}$ mm.

and

$$h_d = m(1.2 - k_c)$$

The centre distance between the gears, on condition that the increase in this distance is equal to the sum of the increases in the centre distances between the gear blanks and the generating tools required to obtain the corrected tooth profiles, can be calculated from the formula:

$$S = P.D. + \sum k_c m$$

The modified teeth obtained by this correction are sharper than ordinary teeth, especially if the number of teeth is small. If the width at the top of the modified teeth is less than 0.1 *m* (module), the height of the teeth should be reduced (by reducing the addendum circle) until a width of 0.1 *m* (module) is obtained.

(E) SCREW PUMPS

The need for pumps that combine the main favourable features of reciprocating and centrifugal pumps, especially in respect to uniform delivery, dry suction, high efficiency at high speeds and small overall size, led to the development of screw pumps. These pumps can handle any liquids and gases without mechanical impurities and create pressures up to 200 kg per sq cm. The rotors may run at speeds in the order of 10,000 rpm and the pump capacities range from 3 to 300 cu m per hour.

Screw pumps may be of the airtight and nonairtight types; they may have either two or three rotors. In the airtight pumps the admission and discharge chambers are separated from each other by airtight meshing of the rotors. Airtight separation of the chambers is absent in the nonairtight type. The most widely used type is the airtight triple-rotor screw pump with cycloidal engagement and with double- or triple-thread screws.

An airtight screw pump for handling oil is shown in Fig. 89. It delivers oil at a pressure of 20 kg per sq cm and has a capacity of 12.6 cu m per sec at a speed of 1,500 rpm. Cast iron cover *2* with the admission connection *13* is secured to the cast iron casing *1* which is provided with the discharge connection. Casing *1* contains the cast iron rotor housing *3* with a port *12* through which oil enters the admission chamber. This keeps the rotor housing full of oil even when the pump is standing idle. The inner surface of the rotor housing is lined with babbitt.

Housing *3* contains the driving, or power, rotor *4* meshing with the driven, or idler, rotors *5*. Sleeves *7* and *8* are fitted on the cylindrical ends of the rotors to take up the axial forces due to the difference in pressure in the discharge and admission chambers. These sleeves bear against the bronze cover *6* of the rotor housing and are fixed by special lugs. The spaces between the end faces of the rotors and the internal surfaces of the sleeves are connected with the discharge chamber through holes drilled through the rotors. This measure provides partial hydraulic balancing of the rotors.

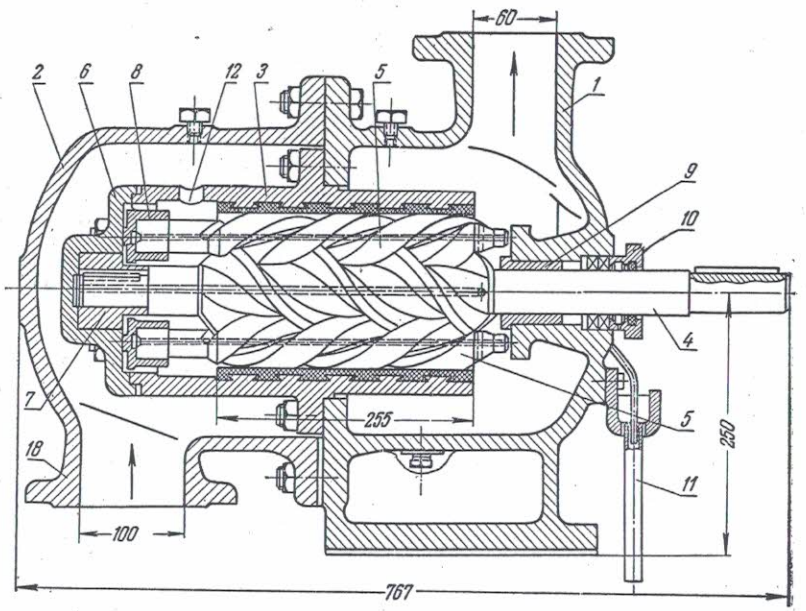


Fig. 89.

The sleeves are lubricated through radial grooves in their end surfaces. The driving shaft is sealed by a stuffing box consisting of bushing *9*, packing and gland *10*. Oil leakage is drained through pipe *11*. In addition to the discharge connection, the discharge chamber has two other connections for mounting the bypass and relief valves.

The liquid being pumped enters the admission chamber and fills the thread grooves, or spaces. When the rotors have turned through a certain angle this volume of liquid is sealed away from the admission chamber and is pushed axially by the thread of the rotor into the discharge chamber.

Therefore, with the rotors rotating uniformly, the liquid also travels uniformly and in a straight line along the rotor axis as does a nut along a screw.

In cross section, the pump rotors have the form of three meshing gears of special shape (Fig. 90) which are called generating gears. The surfaces of the rotors may be considered to be surfaces generated by the helical motion of the generating gears.

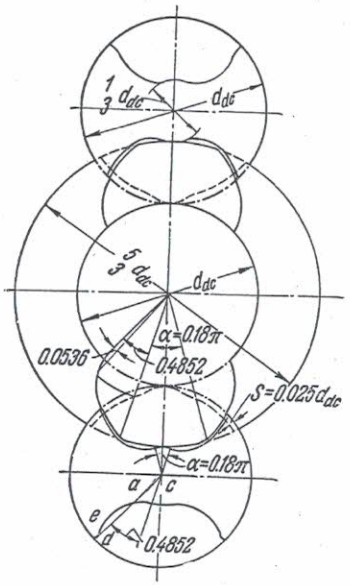


Fig. 90

The diameter of the dedendum circle of the driving rotor threads (considered here as gears) is

$$d_{dc} = 10\sqrt[3]{\frac{Q_{st}}{0.0691n}} = 24.37\sqrt[3]{\frac{Q_{st}}{n}}\ \text{cm} \quad (159)$$

where Q_{st} = theoretical capacity of the pump, litres per sec

n = speed of the driving rotor, rpm.

The diameter obtained from this formula is rounded off to the nearest value which is a multiple of three.

The other principal dimensions of the working parts of a pump with cycloidal engagement are found on the basis of d_{dc} from the relations listed in Table 25.

Table 25

Dimension	Value
Addendum circle diameter of the driving rotor	$\frac{5}{3}d_{dc}$
Dedendum circle diameter of the driving rotor	d_{dc}
Addendum circle diameter of the driven rotors	d_{dc}
Dedendum circle diameter of the driven rotors	$\frac{1}{3}d_{dc}$
Length of the radial land on the driven rotors	$S_0=0.025\ d_{dc}$
Angle $\alpha^\circ=0.18\pi=0.56549$	$\alpha^\circ=32°24'$
Thread lead	$t=\frac{10}{3}d_{dc}$
Minimum length of the thread	$h=5\ d_{dc}$

Data compiled by the Leningrad Engineering Plant show that pumps designed on the basis of this table have the most favourable features in respect to:

(1) maximum useful section of the rotor housing;
(2) maximum rotor speed;
(3) ample strength and rigidity of the driven rotors;
(4) relieving the driven rotors of the motor torque.

The actual capacity of a triple-rotor screw pump with cycloidal engagement is determined from the formula:

$$Q_s = \frac{Ftn}{60}\eta_v \frac{1.24307\times10}{60\times3}\eta_v n d_{dc}^3 = 0.0691 d_{dc}^3 \eta_v n \quad (160)$$

where F = 1.24307 d_{dc}^2=effective cross-sectional area of the rotor thread grooves

η_v = 0.85 to 0.99=volumetric efficiency

$t = \frac{10}{3}d_{dc}$=lead of the rotor thread

n = rotor speed, rpm.

The volumetric capacity per second computed by this formula will be in units corresponding to the one taken for the pitch diameter *P. D.*

The liquid velocity in the suction line is taken in the range from 0.6 to 1.2 m per sec, and in the discharge line—from 2 to 2.5 m per sec.

The maximum velocity of axial liquid flow in the rotor housing should not exceed 5.5 m per sec. This velocity serves as a basis for determining the minimum permissible size of the rotors and their maximum permissible speed. The actual axial velocity of the liquid can be expressed by the equation

$$v_{ax} = \frac{Q_{st}}{10^3 F} = \frac{Q_{st}\,10^4}{10^3\times1.2431 d_{dc}^2} = 8.04\frac{Q_{st}}{d_{dc}^2}\ \text{m per sec} \quad (161)$$

or

$$v_{ax} = \frac{nt}{60} = \frac{10nd_{dc}}{60\times3\times100} = \frac{d_{dc}n}{1{,}800}\ \text{m per sec} \quad (162)$$

where d_{dc} and t are in cm

Q_{st}= theoretical capacity, litres per sec.

The minimum permissible diameter $d_{dc\,min}$ of the dedendum circle on the driving rotor is found from equation (161) for v_{ax}=5.5 m per sec

$$d_{dcmin} = 2.84\sqrt{\frac{Q_{st}}{v_{ax}}} = 1.21\sqrt{Q_{st}}\ \text{cm}$$

The maximum permissible rotor speed is found from equation (162) and the preceding equation for a liquid velocity v_{ax}=5.5 m per sec

$$n_{max}=\frac{1{,}800v_{ax}}{d_{dc\,min}}=\frac{1{,}800v_{ax}\sqrt{v_{ax}}}{2.84\sqrt{Q_{st}}}=634\frac{v_{ax}^{3/2}}{\sqrt{Q_{st}}}=634\frac{5.5^{3/2}}{\sqrt{Q_{st}}}=\frac{8{,}175}{\sqrt{Q_{st}}}\ \text{rpm} \tag{163}$$

It follows from the above that upon an increase in capacity, the permissible speed of the rotors is reduced.

The working length of the rotor for z_{thd} leads of thread is

$$L=tz_{thd}=\frac{10}{3}d_{dc}z_{thd}\ \text{cm} \tag{164}$$

where z_{thd} = number of leads of thread on the working length of the rotor.

The rotors are usually double- or triple-thread screws. The number of leads of thread z_{thd} required on the rotor and its working length increase with the discharge pressure of the pump.

Table 26 lists the required number of leads, z_{thd}, in accordance with the pressure developed by the pump.

Table 26

Pressure developed by the pump	Number of leads of thread, z_{thd}
Low pressure — up to 15 kg per sq cm . . .	1.5
Medium pressure — up to 60 kg per sq cm . .	3
High pressure — up to 175 kg per sq cm . . .	6

The power required to drive a screw pump is equal to

$$N=\frac{Q_s(p_d-p_s)}{10.2\eta_m\eta_v}=\frac{0.6774(p_d-p_s)nd^3_{dc}}{10^5\eta_m\eta_v}\ \text{kW} \tag{165}$$

where p_d and p_s = discharge and suction pressures, respectively, in kg per sq cm
η_m = mechanical efficiency of the pump
d_{dc} = dedendum circle diameter of the generating gears, cm
Q_s = actual pump capacity, litres per sec.

Characteristic curves of the pump described above are shown in Fig. 91. Another type of screw pump, designed with liquid admission to the rotors from both ends, has admission chambers arranged at opposite ends of the pump. This design enables the rotor speed to be increased and balances the rotors against axial pressure. In pumps of this type with cycloidal engagement the maximum permissible

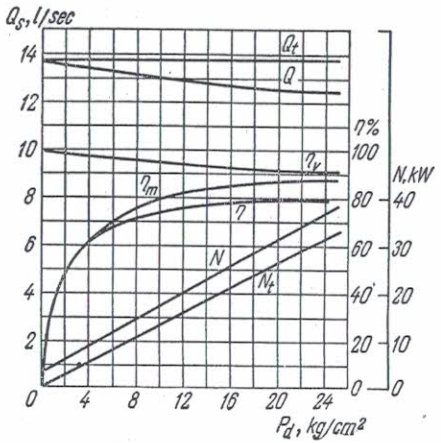


Fig. 91.

rotor speed is found from the formula:

$$n_{max}=\frac{8{,}175}{\sqrt{\frac{Q_{st}}{2}}}\ \text{rpm} \tag{166}$$

The maximum vacuum that can be created by screw pumps with cycloidal engagement when handling liquids of various viscosities is indicated by the curves in Fig. 92.

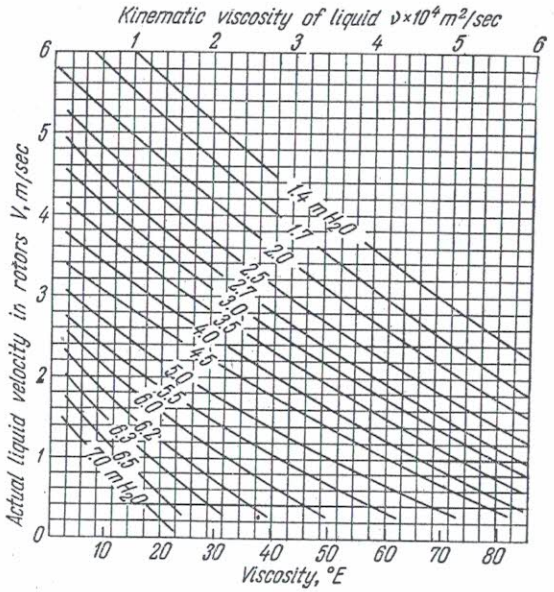
In the nonairtight types of screw pumps the rotors have square threads. The pressure that can be developed by screw pumps depends upon the clearances between the rotors and the rotor housing. Screw pumps handling viscous liquids can operate for years without repairs.

The capacity of screw pumps with square thread rotors can be calculated with sufficient accuracy for practical purposes from the formula:

$$Q=60\eta_v k_i\frac{\pi}{4}(D^2_{mj}-D^2_{mn})\frac{t}{2z_{thd}}-n=$$

$$=7.5\eta_v k_i(D^2_{mj}-D^2_{mn})\frac{\pi t}{z_{thd}}\ \text{cu m per hour} \tag{167}$$

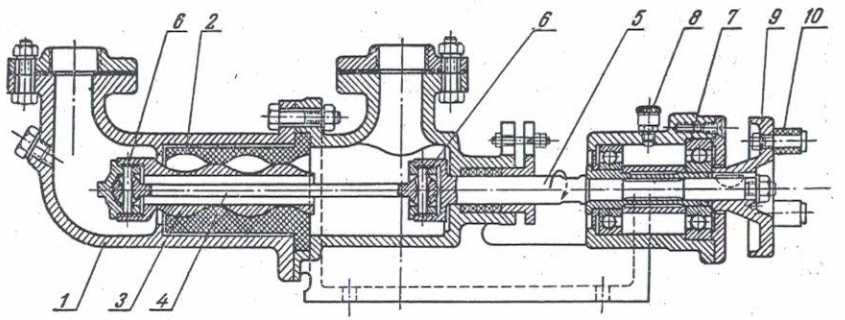
where k_i = number of driven (idler) rotors
t = pitch of the square thread on the rotors, m
z_{thd} = number of threads (starts) on the rotor
n = rotor speed, rpm
D_{mj} = major diameter of the square thread, m
D_{mn} = minor diameter of the square thread, m.



F i g. 92.

What may be considered as a special type of single-rotor screw pump is shown in Fig. 93. Here, the metal casing *1* contains a rubber helix *2* in which a metal rotor *3* rotates. The rotor is a single-threaded worm while the stationary helix in which it rotates is double-threaded and has a lead equal to twice the pitch of the rotor. The rotor is linked to the intermediate shafts *4* and *5* through the universal joints *6* (similar to a cardan shaft arrangement) enabling the rotor to roll with an eccentric motion in the stationary helix in accordance with the shapes of the rotor and helix. When the rotor is driven it rotates about its axis and, at the same time its axis describes a small circle in the opposite direction. As the rotor rolls in the stationary double helix, it maintains a sealed contact throughout its length. Its eccentric rolling motion produces an advancing positive displacement

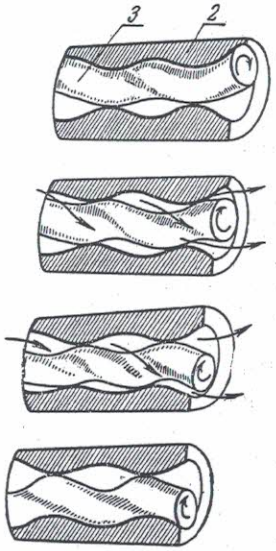
from the suction to the discharge of the pump. The principle of operation is clear from the four positions of the rotor shown in Fig. 94.



F i g. 93.

The intermediate shaft *5* runs in ball bearings *7* and is sealed with an ordinary gland packing. Lubricant is supplied to the ball bearings through the grease cup *8*.

The pump shaft is linked to the drive motor or engine through flange *9* carrying the pins *10* on which rubber bushings are fitted. The casing is designed so that it always contains liquid when the pump stands idle; starting the pump dry may lead to destruction of the rubber helix. Capacities of these pumps range from 4 to 10 cu m per sec at rotor speeds as high as 2,800 rpm. The suction lift is 8 mH_2O while heads up to 40 mH_2O are created.



F i g. 94.

The favourable features of screw pumps include:

(1) high overall efficiency (up to 70 or 80 per cent) at a volumetric efficiency $\eta_v = 0.91$ to 0.995;

(2) only a slight increase in slip with an increase in the discharge pressure;

(3) small overall size and low weight as a result of the high rotor speeds employed;

(4) high pressures created by the pumps (up to 200 kg per sq cm);

(5) slight wear of the working parts in conjunction with a mechanical efficiency from 0.76 to 0.86 and even higher;

(6) straight uniform motion of the liquid excludes pulsation and agitation;

(7) capability of operating with the casing in any position in respect to the horizontal;
(8) good suction capacity;
(9) slight vibration of the foundation as a result of effective balancing of the moving parts.

The wide application of screw pumps is due to these main features.

The suction lift of screw pumps is determined by the formula:

$$\frac{P_s}{\gamma}=\frac{P_a}{\gamma}-\left[z_s+\frac{v_s^2}{2g}+\sum\zeta_i\frac{v_i^2}{2g}\right]\geqslant 6\text{mH}_2\text{O} \qquad (168)$$

where z_s = geodetic lift, m
v_s = liquid velocity in the admission chamber, m per sec
ζ_i = summary coefficients of hydraulic resistance in the sections of the suction line with the corresponding liquid velocities, v_i, m per sec.

(F) WATER-RING PUMPS

The principle of a water-ring, or liquid-ring, pump is shown schematically in Fig. 95. The working chamber of the pump consists of the circular casing *1* in which the rotor *2* is mounted eccentrically. The rotor is keyed on the pump shaft *3*. The action is as follows. First the initial charge of liquid is poured into the casing. When the rotor begins to revolve it drives around with it the liquid caught between its radial blades. After the rotor reaches full speed, the centrifugal force acting on the liquid causes it to form a continuous ring against the casing. The crescent-shaped ports *4* and *5* in the casing covers communicate with the admission and discharge connections of the pump. If rotor *2* revolves in the clockwise direction, admission will be through port *4* and discharge through port *5*. This takes place because the volume of the cells, or pockets, formed by two adjacent blades, the inner surface of the liquid ring and the sides of the casing, increases at the right side of the working chamber and diminishes at the left side.

If the liquid ring fills the casing to a point where its inner surface just contacts the rotor hub, and if the number of rotor blades is denoted as z, their mean thickness as S, their width as b and the rotor speed in rpm as n, then the flow rate of auxiliary water per second will equal

$$Q_{sa}=\left[\frac{\pi}{4}(D_r^2-D_h^2)-zl_1S\right]b\frac{n}{60}\ \text{cu m per sec} \qquad (169)$$

If we take the slip into consideration by introducing the volumetric efficiency η_v and denote the length of the blade submerged

to the minimum extent in the liquid as a, the length of the rotor blades as l_1, expressing all linear dimensions in m, then the actual hourly capacity of the pump will be

$$Q\equiv 60\eta_v\left\{\frac{\pi}{4}\left[(D_r-a)^2-D_h^2\right]-z(l_1-a)S\right\}bn \qquad (170)$$

cu m per hour

Water-ring pumps are most frequently used as vacuum pumps with capacities up to 1,800 cu m per hour at rotor speeds up to 1,430

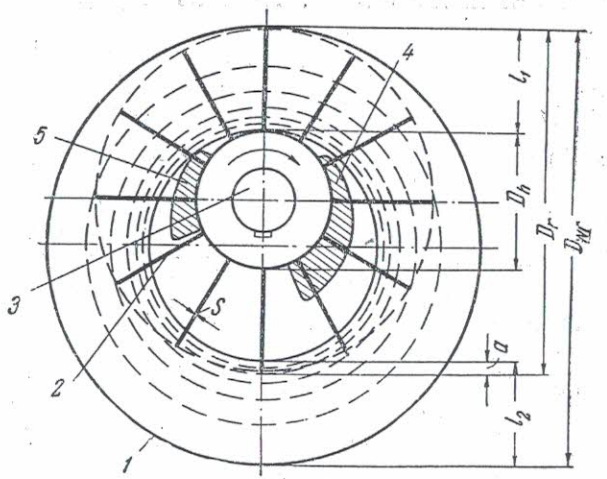


Fig. 95.

rpm. Since no precise fitting of the parts is needed to create the vacuum, water-ring pumps are easy to manufacture, install, maintain and operate.

Vacuum pumps of the rotary type can create a vacuum of 99.5 per cent of the theoretical possible value if the parts are precisely manufactured and fitted.

The required capacity of a vacuum pump that is to serve a vessel with a volume V_{ves} of air is found from the formula:

$$Q_{vac}=\frac{V_{ves}}{\tau}\ \text{cu m per min} \qquad (171)$$

where τ = time required to create a vacuum, min.

If the vacuum pump is used as an air-exhausting pump to create a vacuum in a centrifugal pump, then the value τ is taken to be:
(a) not over 0.5 min for fire pumps,
(b) not over 0.5 to 1 min for other pumps.

(G) TURBINE PUMPS

The action of a turbine pump (also called vortex, periphery and regenerative pumps) is based on imparting kinetic and potential energy to the liquid in the spaces between the impeller vanes, and on continuous conversion of kinetic into potential energy as the liquid passes from one space between the vanes to the next.

Let us consider the turbine pump illustrated in Fig. 96. The spaces between the vanes, or cells as we shall call them, may be formed by either radial or inclined vanes. The cells may extend across the

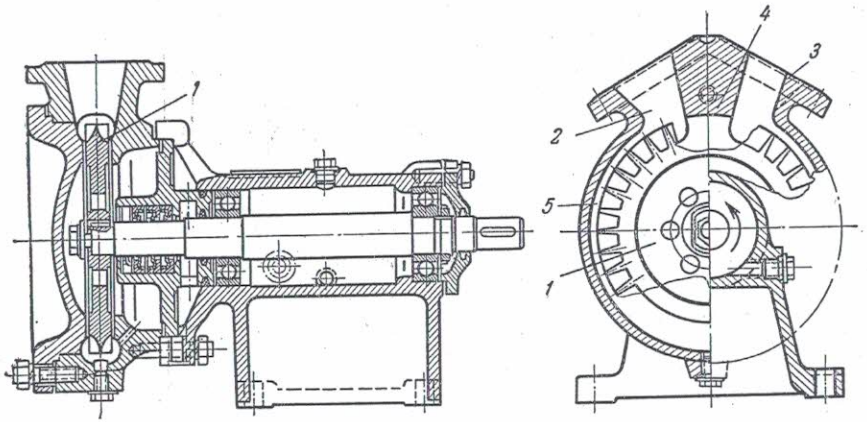


Fig. 96.

whole width of the impeller or may be divided by a central partition into two cells, one on each side, as shown in Figs. 96 and 97.

Upon rotation of the impeller *1* in the clockwise direction, the liquid, admitted through connection *2*, is picked up by the vanes and carried along the annular channel *5* to the discharge connection *3* from where it is delivered into the discharge line. The admission and discharge are separated by the seal (partition) *4*. In its travel around the pump each portion of liquid describes a complex path.

Caught up by the impeller at point *A* (Fig. 97) the portion of liquid participates in rotary motion in the direction of impeller rotation, relative motion along the surfaces of the cell and absolute motion. The direction of the absolute motion is determined by the direction of the absolute velocity *c* which is obtained as the geometric sum of the relative velocity w and the rotary velocity u.

During this motion the portion of liquid acquires potential and kinetic energy with which it is thrown out into channel *5* of casing *4* at a certain angle forward in the direction of impeller revolution

(point *B*). In the channel the kinetic energy is partly converted into potential energy, and the portion of liquid is caught up by the next vane (point *C*) after describing a helical path in the channel. Thus, in its progress towards the discharge connection, the portion of liquid acquires more and more energy. The infinite number of portions of liquid form a helical flow of increasing static head as it approaches the discharge connection. The casing of a turbine pump must be filled with liquid to ensure its self-priming properties.

Fig. 98 illustrates the path of the liquid in respect to the stationary casing.

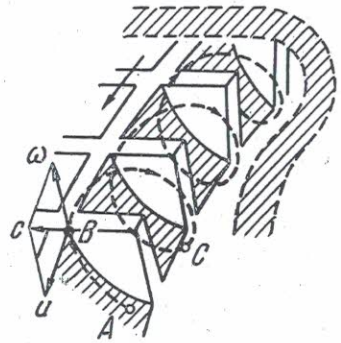


Fig. 97.

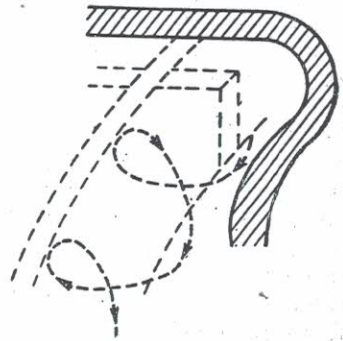


Fig. 98.

Turbine pumps find extensive application, due to their simple construction, as vacuum pumps in installations where the vacuum required does not exceed 4 or 5 mH_2O.

In some cases, turbine pumps supplant reciprocating pumps because of the capability to handle polluted liquids and liquids of both low and high viscosity, as well as the high pressures they can develop. The main drawback of turbine pumps is their low overall efficiency which ranges from 30 to 40 per cent.

(H) OPERATION OF ROTARY PUMPS

Starting up and stopping a rotary pump and the attention it requires during operation are the same as for reciprocating pumps.

Some of the troubles that may be encountered in operation and their most frequent causes are:

1. The pump does not reach its full capacity. This may be due to the following causes: air leaks develop in the suction line, the liquid being handled has too high a temperature, the rotor parts and packing are worn, valves in the line are not fully opened, the filters are clogged, or the rotor speed is too low.

2. No liquid is delivered. This may be the result of one of the preceding causes and also because the valves in the line are closed, the filters are badly clogged, the rotor shows heavy wear or is incorrectly installed in reference to the free surface of the liquid being pumped (the static lift is too high).

3. The pump overheats during operation. The temperature of various parts may rise above the normal value because of the excessively high temperature of the liquid, excessive or nonuniform compression of the packing which leads to seizing of the pump shaft, hardening of the packing, misalignment of the shafts and rotors of the pump, compression of the liquid in the pump casing, failures in lubrication of the friction surfaces, or interference between the rotors and the internal parts of the pump casing.

4. Severe vibrations of the casing are observed. This is commonly due to the following: foundation bolts are loose, the motive unit and pump shafts are misaligned, excessive play is observed in the coupling that links the motive unit to the pump, the pump rotors are poorly balanced, the pump rotors have got out of alignment, the rotors interfere with the casing because the shaft is bent or the bearings are excessively worn, the rotor speed approaches the critical value, or foreign substances have got into the casing.

The causes of abnormal pump operation impair performance and increase wear and maintenance expenditures. The pump should not be allowed to operate even for a short time when it is out of order as this may result in a serious breakdown.

(I) HAND PUMPS

Hand pumps are used as the main pumping units of sailing ships and small engine-driven vessels. In large ships they are used mainly for auxiliary purposes such as the pumping of fresh and sea water, liquid fuel and oil.

Hand pumps are also used in large ships to drain compartments into which water leakage is insignificant and does not require systematic attention and also for draining water from places in which the installation of a branch of the drainage system proves difficult.

Hand pumps are also mounted on auxiliary boilers in cases permitted by the regulations of the U.S.S.R. Shipping Register.

Hand pumps may be of the piston or vane types, either of which may by single- or double-acting.

The capacity of hand pumps may be as high as 13 cu m per hour, their suction lift up to 4 or 5 mH_2O and discharge head up to 12 kg per sq cm.

The principal designs of hand piston pumps are shown schematically in Fig. 99. Those shown in Fig. 99*a* and *b* are double-acting

while that in Fig. 99*c* is a single-acting pump with a differential piston. The designs are so simple that further explanations will hardly be required.

The construction of a double-acting hand piston pump is illustrated in Fig. 100. The casing *1* with the suction connection *2* and the discharge connection *3* contains the cylinder *11* in which the piston *6* reciprocates. The piston is driven by the roller at the short end *10* of lever *9* which rocks on axle *8*. The roller enters a slot in the piston, and the lever is rocked by means of the handle *12*. Each end of the cylinder has its inlet *4* and discharge *5* valves. The pistons are sealed with snap rings *7*.

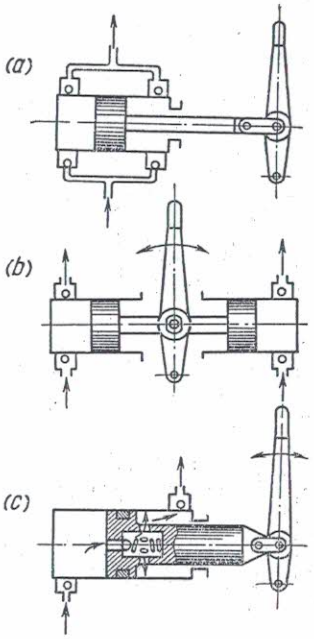


Fig. 99.

When the handle is rocked in the counterclockwise direction, the pistons travel from left to right so that the liquid is discharged from the right end of the cylinder and drawn into the left end. These functions will be reversed on the clockwise stroke of the handle. Thus, in two strokes the pump will discharge a volume of liquid equal to two volumes of the cylinder.

The capacity of a hand pump and its principal dimensions are computed from the same formulas that are employed for reciprocating power pumps.

A more compact pump, but one with a lower volumetric efficiency and requiring more exact fitting of its parts, is the hand vane pump.

Hand vane pumps may be either single- or double-acting, and they are used for the same purposes as hand piston pumps. A drawback of these pumps is their low volumetric efficiency but, on the other hand, they present less difficulty in manufacture.

A single-acting hand vane pump is illustrated in Fig. 101. The casing *1* contains the vane *2* which carries the flap valves *3* and *4* and is mounted on the shaft *5* together with handle *6*. The concave upper end of partition *7*, equipped with the inlet valves *8* and *9*, is fitted to the hub of the vane.

As the handle is rocked back and forth, as indicated by the arrows, admission and discharge takes place in the corresponding chambers of the casing. In the position shown in Fig. 101 and with the handle moving in the clockwise direction, liquid is being drawn into the

lower left chamber *11* and discharged from the lower right chamber *10*.

In one stroke of the handle this pump delivers an amount of liquid equal to the working volume of the pump casing.

The double-acting pump shown in Fig. 102 discharges approximately twice as much liquid to each oscillating movement of the handle than does the pump shown in Fig. 101.

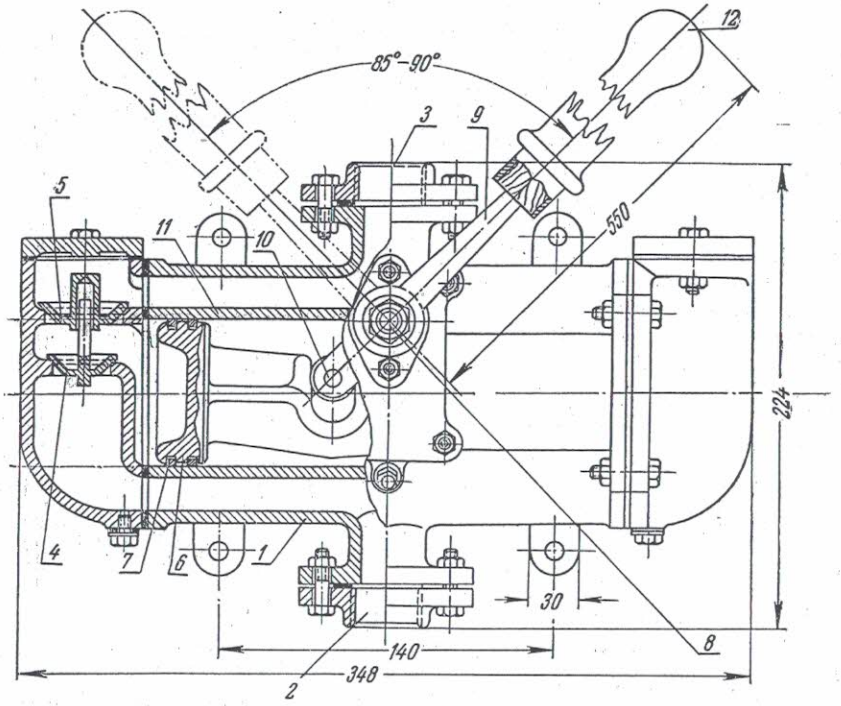


Fig. 100.

Higher displacement is achieved in this case by providing passages in the vane hub that connect the diagonally opposed chambers formed by the vane, casing and valve chest walls. The casing *1* houses the valve chests with inlet valves *2* and discharge valves *3*, and the vane *4* mounted on its shaft *5*. Handle *6* secured on the end of shaft *5* is used to drive the vane.

When the handle is rocked in the clockwise direction, liquid is drawn into the lower left *9* and upper right *11* chambers in the casing from the suction line through the flanged connection *6*. At the same time, liquid is discharged from the lower right *8* and upper left *10* chambers through the flanged connection *7* into the discharge line.

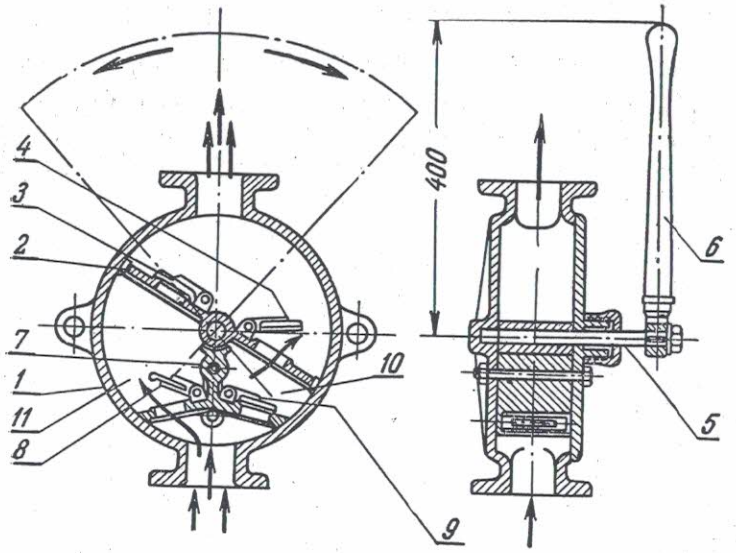


Fig. 101.

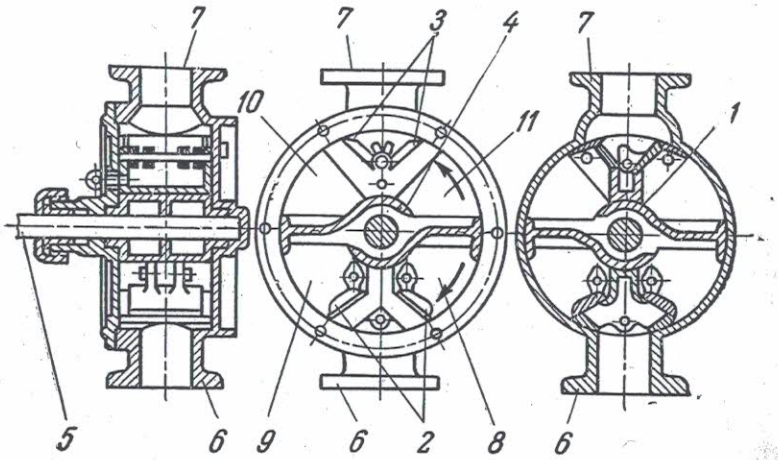


Fig. 102.

D. CENTRIFUGAL AND PROPELLER PUMPS

1-20. Principles and Classification of Centrifugal Pumps

The principle of the centrifugal pump is illustrated in Fig. 103. The split casing, having a volute chamber *5*, contains the impeller *6* that consists of the circular disks, or shrouds, *3* and *2*, between

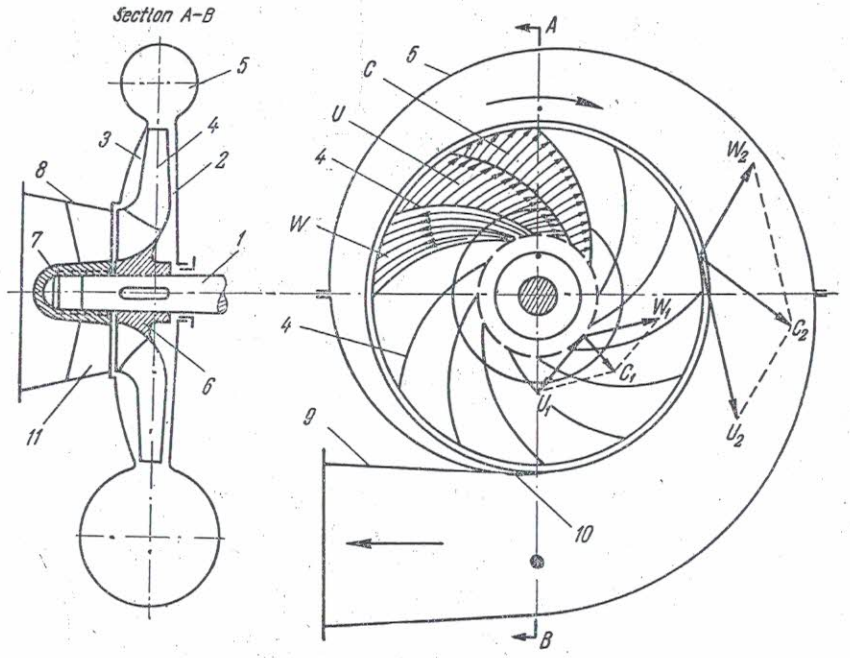


Fig. 103.

which the vanes *4* are secured. The impeller hub is keyed on shaft *1* where it is held by streamlined nut *7*. Shaft *1* is driven by the motive unit.

Liquid is admitted to the impeller through inlet *8*. The liquid passes through the impeller into the volute chamber *5* from where it is discharged through the conical outlet branch *9* into the discharge line. The volute begins at the narrowest part *10*, called the tongue, and serves to collect the liquid discharged from the impeller vanes, converting the dynamic flow, in some degree, into static flow.

A centrifugal pump will operate normally if, when started, the cavities of the casing, spaces between the impeller vanes and the suction line have in some manner been filled with liquid. Upon

rotation of the impeller, the liquid in the spaces between the vanes also begins to rotate due to the mechanical action of the vanes, and the particles of liquid acquire a peripheral velocity u. Velocity u is tangent to a circle described from the centre of the impeller through any given point. The centrifugal force generated in the mass of liquid between the vanes due to its peripheral velocity u develops a pressure at the outer diameter of the impeller. If this pressure is sufficient to overcome the counter-pressure in the discharge line, the liquid in the intervane spaces begins to flow and its particles travel in the channels between the vanes from the centre to the periphery with a relative velocity w. The latter is directed tangent to the curve of the vane. Therefore, the absolute velocity c of movement of the particles of liquid in the impeller channels is the geometric sum of the peripheral velocity u and the relative velocity w. In the ideal case, if we deal with an inviscid fluid and the impeller has an infinite number of blades, all the particles of liquid will travel along identical trajectories. Sometimes, inlet guide vanes *11* are installed before the impeller to admit the particles of liquid along a tangent to the surfaces of the impeller vanes. The velocities u, w and c are acquired by the liquid because the energy developed by the motive unit is imparted to the flow of liquid as it passes through the rotating impeller. The amount of energy imparted to the flow by the motive unit is characterized by the increase in flow velocity from u_1, w_1 and c_1 at the inlet circumference of the impeller to u_2, w_2 and c_2 at the outlet circumference. The absolute velocity c_2 of the liquid leaving the impeller is always higher than that permitted in the discharge line and, therefore, the dynamic head of the liquid stream is converted into static head. This is known as recuperation. According to Bernoulli's equation this conversion may be achieved by reducing the velocity of the liquid stream. To this end, the impeller discharges the liquid into a progressively expanding passageway in the form of a volute whose cross section gradually increases towards the outlet so as to obtain the required reduction in velocity. This action of the volute is multiplied by the conical outlet branch *9*. The intervane channels, being freed of the liquid discharged into the volute, are filled with liquid admitted to the impeller from the suction line due to the pressure of the atmosphere on the free surface of the volume of liquid being pumped. Suction is accomplished in a centrifugal pump in this manner. The practically uniform rotation of the impeller sets up a steady flow of liquid in the piping of the pump installation.

Centrifugal pumps are classified

1. According to their capacity:

(a) Low-capacity pumps—up to 20 cu m per h.

(b) Medium-capacity pumps—from 20 to 60 cu m per h.

(c) High-capacity pumps—over 60 cu m per h.

2. According to the pressure developed:
(a) Low-pressure pumps—up to 5 kg per sq cm.
(b) Medium-pressure pumps—from 5 to 50 kg per sq cm.
(c) High-pressure pumps—over 50 kg per sq cm.
3. According to their speed:

Pump speeds are characterized by the specific rotational speed, or speed factor, of the impeller.

The specific velocity, or speed factor, is defined as the impeller speed of a model pump which, being geometrically similar to the given pump, develops a head of one metre water column at a discharge of 75 litres per sec having a power consumption of one metric hp at highest efficiency.

4. According to their specific velocity, n_{s1}:
(a) Low-speed pumps—n_{s1}=40 to 80.
(b) Moderate-speed pumps—n_{s1}=80 to 150.
(c) High-speed pumps—n_{s1}=150 to 300.
(d) Mixed-flow pumps—n_{s1}=300 to 600.*
5. According to the number of impellers and stages.
(a) Single-impeller pumps having one impeller and in which the pressure is generated in a single stage (Fig. 103).
(b) Multistage pumps in which the passages of several impellers, mounted in a single casing, are connected in series.
(c) Multi-impeller pumps in which the passages of several impellers, mounted in a single casing, are connected in parallel.
(d) Multi-impeller multistage pumps in which the passages are connected for a combination of both parallel and series operation.
6. According to the intake to the impeller:
(a) Single admission, in which the liquid enters each impeller on one side or face only.
(b) Double admission, in which the liquid enters the impeller on both sides in opposite directions.
7. According to the design of the casing:
(a) Single casing, in which the casing is split only once along a horizontal plane passing through the pump axis.
(b) Sectional pumps, in which the casing comprises several sections, each section with its impeller constituting a stage of the pump. The casings of the sections are held together as an integral unit by bolts.
8. According to the disposition of the impeller axis:
(a) Vertical-shaft pumps.
(b) Horizontal-shaft pumps.
9. According to the kind of liquid handled:
(a) Water pump.

* A specific velocity over 600 corresponds to an axial-flow, or propeller, pump.

(b) Petroleum pump, etc.
10. According to the method of drive:
(a) Power pumps, linked to the motive unit through some type of transmission (reducing gear, hydraulic drive, etc.).
(b) Direct-coupled pump which is linked directly to the shaft of the motive unit.
11. According to the suction lift capacity:
(a) Self-priming pumps, equipped with a device for creating a vacuum in the suction line when the pump is started.
(b) Non-priming pumps, not having such a device.

Centrifugal pumps are usually designed with one impeller if the latter is sufficient to provide the required head and output (Figs. 103, 106 and 107).

If the required head cannot be generated by a single impeller, several impellers are arranged to operate in series (multistage pumps, Figs. 110 and 109).

If the required capacity cannot be provided at the head developed by one impeller, several impellers are arranged to operate in parallel (multi-impeller pumps) or double admission pumps are used (Fig. 108).

A combination of high head and high capacity may be achieved by arranging the impellers for combined series and parallel operation.

A selection of the number of stages required in a pump may be made on the basis of the specific velocity.

The specific velocity of a single-stage pump is expressed by the formula

$$n_s = 3.65 \frac{n\sqrt{Q_s}}{\sqrt[4]{H^3}} \qquad (172)$$

where n=pump impeller speed, rpm
Q_s=capacity of the pump, cu m per sec
H=head developed by the pump, mH_2O.

In accordance with equation (172) the specific velocity of one stage of a multistage pump will be

$$n_{s1} = 3.65 \frac{n\sqrt{Q_s}}{\sqrt[4]{H_1^3}} \qquad (173)$$

where H_1= head developed by one stage, mH_2O.

It is evident from Fig. 104 that the ratio of the main dimensions of the impeller and its specific velocity vary with its rotational speed.

Specific velocities that are not within the limits indicated in the table (Fig. 104) are characteristic of multistage and multi-impeller pumps.

The specific velocity is a value that links together the capacity, head and power consumption of a pump, all of which depend upon the principal dimensions and velocity of liquid flow in the impeller. Thus, the specific velocity is one of the main values used in determining the most efficient principal dimensions of a pump.

Therefore, the type of pump most suited for any given capacities and heads can be determined on the basis of the calculated specific velocity n_s.

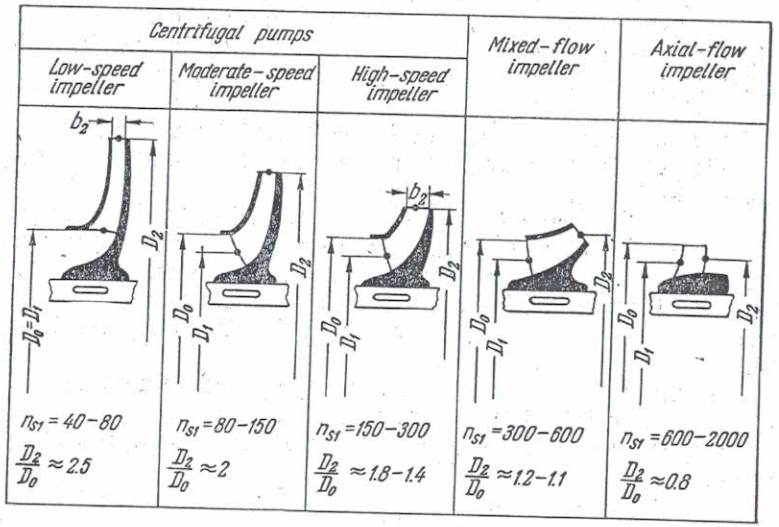


Fig. 104.

The specific velocities indicated in Fig. 104 are characteristic of single-impeller pumps of the corresponding rotational speeds.

Values of n_s that are below the lower limits indicated in the table refer to multistage centrifugal pumps; in these cases, a reciprocating pump may also be employed if it is suitable from other considerations.

The specific velocity of one stage of a multistage pump is

$$n_{s1} = 3.65 \frac{n\sqrt{Q_s}}{\sqrt[4]{H_1^3}} = 3.65 \frac{n\sqrt{Q_s}}{\sqrt[4]{\left(\frac{H}{i}\right)^3}} = 3.65 \frac{n\sqrt{Q_s}}{\sqrt[4]{H^3}} \sqrt[4]{i^3} = n_s \sqrt[4]{i^3} \quad (174)$$

where i = number of stages of pressure
H_1 = head developed by one impeller, mH_2O
H = head developed by the pump, mH_2O.

From equation (174) we obtain

$$i = \sqrt[3]{\frac{n_{s1}^4}{n_s^4}} = \sqrt[3]{\left(\frac{n_{s1}}{n_s}\right)^4} \quad (175)$$

The use of multistage pumps is justified by their higher efficiency resulting from the smaller impeller diameters and, consequently, lower losses from the friction of the liquid on the impeller shrouds.

Specific velocities above the upper limits of the tabular values are inherent in multi-impeller centrifugal pumps arranged for parallel operation of the impellers or for double admission pumps. A mixed-flow pump can also be applied in this case.

The specific velocity for one impeller of a multi-impeller pump is

$$n_{s1} = 3.65 \frac{n\sqrt{Q_{s1}}}{\sqrt[4]{H^3}} = 3.65 \frac{n\sqrt{\frac{Q_s}{k_k}}}{\sqrt[4]{H^3}} = 3.65 \frac{n\sqrt{Q_s}}{\sqrt[4]{H^3}} \frac{1}{\sqrt{k_k}} = \frac{n_s}{\sqrt{k_k}} \quad (176)$$

where Q_{s1} = discharge through an impeller with single admission, cu m per sec
k_k = number of admissions (entries) to the impellers of the pump.

On the basis of equation (176) the number of admissions is

$$k_k = \left(\frac{n_s}{n_{s1}}\right)^2 \quad (177)$$

The permissible head developed by one impeller is limited and an arrangement of series and parallel operation is resorted to if the required head, H, excludes the use of a single-impeller pump, even if the specific velocity value, n_s (Fig. 104), is such that one could be employed.

In such cases, the specific velocity of one impeller will be

$$n_{s1} = 3.65 \frac{n\sqrt{\frac{Q_s}{k_k}}}{\sqrt[4]{\left(\frac{H}{i}\right)^3}} \quad .65 \frac{n\sqrt{Q_s}}{\sqrt[4]{H}} \frac{\sqrt[4]{i^3}}{\sqrt{k}} = n_s \frac{\sqrt[4]{i^3}}{\sqrt{k}} \quad (178)$$

1-21. Suction Lift of Centrifugal Pumps

A pumping installation with a centrifugal pump located above the level of the liquid being handled is shown schematically in Fig. 105.

Let us consider a case in which suction pipe *4*, casing *5* and the intervane spaces of impeller *16* are filled with air. When the pump

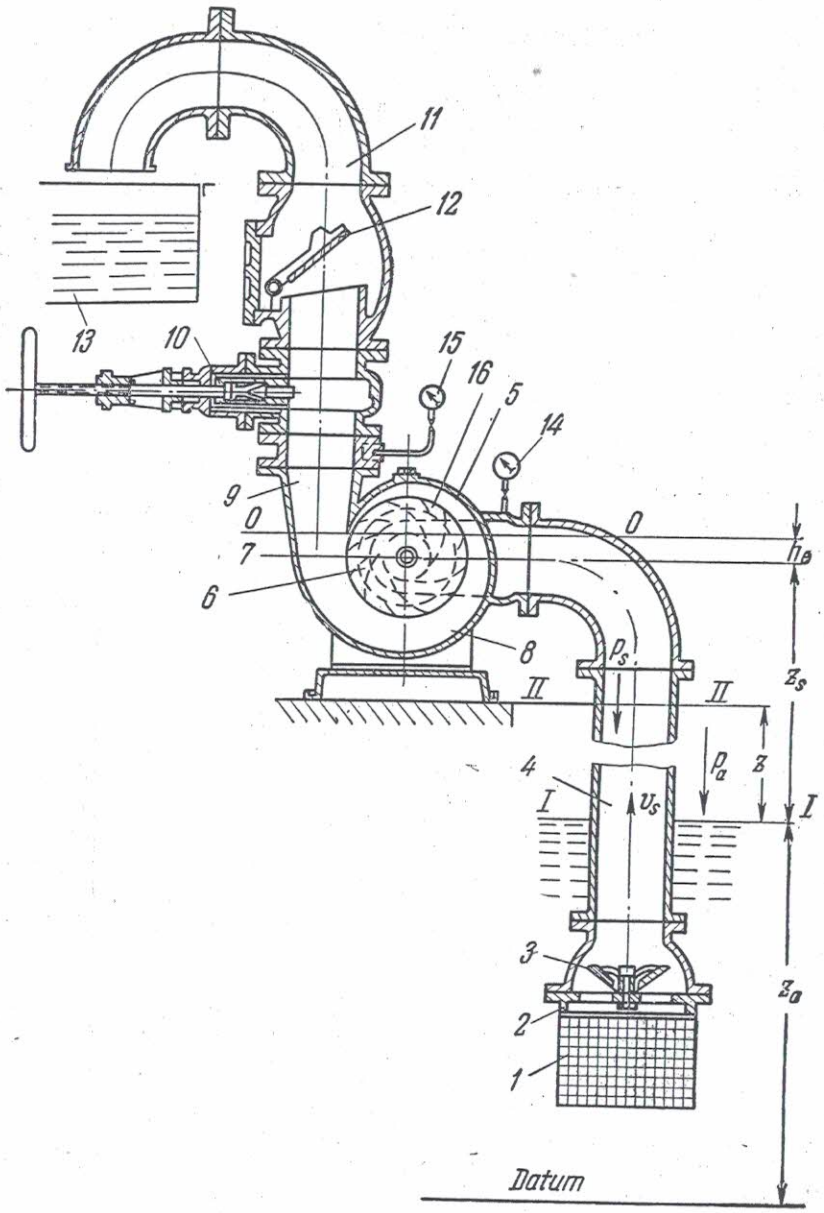


Fig. 105.

is started shaft *7* begins to rotate the impeller. The mechanical action of the rotating vanes *6* discharges the air from between the vanes into volute *8* at a higher pressure, and from there the air passes into the conical outlet *9*. This air is replaced by air drawn in from the suction pipe. A part of the air leaving the impeller leaks back through clearances between the pump casing and the impeller shrouds into the suction line. Moreover, the head of air developed by the impeller is insufficient to overcome the hydraulic resistance of the discharge line which is usually full of liquid. To ensure the removal of air from the suction line, it is necessary to prevent its leakage into the inlet connection after leaving the impeller. It is also necessary to provide peripheral velocities at the outlet circumference of the impeller sufficient to develop a head that could overcome the hydraulic resistances in the discharge line. It proves practically impossible to impart such peripheral speeds to the impeller.

At the present time, air is removed from the pump casing and suction line either by filling (priming) them with the liquid being handled or by means of a vacuum pump connected to the centrifugal pump.

Even though centrifugal pumps are not capable of operating with dry suction, their suction lift in steady operation is higher than that of reciprocating pumps due to the absence of valves and inertia heads of liquid.

The pump can be started subsequently without priming if the casing and suction pipe remain filled with liquid after the pump is stopped. For this purpose, the inlet box *2* with the filtering strainer *1* is equipped with nonreturn flap valves (foot valves) *3*.

The pump is started with the discharge regulating gate valve *10* in the open position. Impeller *16* imparts additional energy to the liquid between the vanes and thereby increases its head. If this head is sufficient to overcome the pressure in the discharge pipe *11*, discharge valve *12* opens and the liquid is delivered to the discharge tank *13*. The degree of vacuum in the suction pipe is indicated by vacuum gauge *14*, and the pressure in the discharge pipe, by pressure gauge *15*.

Since the impeller of a centrifugal pump rotates at a constant angular velocity, a steady flow of liquid is established in the channels of the installation. Therefore, Bernoulli's equation can be applied to each cross section of the flow in the system.

After establishing a datum plane we can write for cross sections *I-I* and *II-II*:

$$z_a + \frac{p_a}{\gamma} + \frac{v_a^2}{2g} = (z_a + z) + \frac{p_s}{\gamma} + \frac{v_s^2}{2g} + h_l \text{ mH}_2\text{O} \qquad (179)$$

where z_a = distance from the free surface of the liquid to the datum, m
p_a = pressure on the free surface of the liquid, kg per sq m
v_a = velocity of the liquid at the free surface in the direction of the inlet strainer, m per sec
z = distance from cross section *II-II* to the free surface of the liquid, m
p_s = pressure of the liquid at cross section *II-II*, kg per sq m
v_s = liquid flow velocity at cross section *II-II*, m per sec
h_l = loss of head in the suction pipe from the strainer to cross section *II-II*, mH_2O.

It follows from equation (179) that the suction head at cross section *II-II* is

$$\frac{P_s}{\gamma}=\frac{P_a}{\gamma}+[z_a-(z_a+z)]-\frac{v_s^2}{2g}\left(1-\frac{v_a^2}{v_s^2}\right)-h_l \quad mH_2O \qquad (180)$$

Assuming that velocity $v_a \approx 0$, equation (180) can be written as

$$\frac{P_s}{\gamma}=\frac{P_a}{\gamma}-\left(z+\frac{v_s^2}{2g}+h_l\right) \quad mH_2O \qquad (181)$$

If the inlet, or entry, velocity of the liquid is denoted as c_e, the head at the inlet circumference of the impeller will be

$$\frac{P_s}{\gamma}=\frac{P_a}{\gamma}-\left(z_s+h_e+\frac{c_e^2}{2g}+h_{lh}\right) mH_2O \qquad (182)$$

where h_{lh} = loss in head in the suction line due to hydraulic resistances, mH_2O
h_e = radius of the impeller inlet, or entry, circumference, m.

According to equation (182) the suction head will decrease with an increase in z_s.

The pressure p_s will not be the same at all points of any given cross section because of the nonuniform distribution of velocity of liquid flow over the cross section in the channels of the system.

At places where the pressure p_s drops to a value at which gases, dissolved in the liquid, are evolved, cavities are formed in the liquid. If these cavities are small, they do not affect normal pump operation, however, increased wear of the parts of the pump installation is observed at the places where they appear.

It is well known that the vapour pressure p_v of a liquid depends upon its temperature, and that the boiling of a liquid at a low pressure occurs at a correspondingly lower temperature.

Consequently, if pressure p_s drops to the value of the vapour pressure of the liquid being pumped at the given temperature, the liquid begins to boil and the intensive vapour generation increases

the volume of the cavities which will now be filled with both gases and vapour.

This may have a detrimental effect on pump operation and, as a first result, may reduce its output.

Because of the nonuniform distribution of the velocity of liquid flow over the cross section at any point in the pumping system, vapour generation may occur in one or several separate zones. In this case, the gases and vapour evolved from the liquid and filling the low-pressure zones are swept by the flowing liquid to regions of higher pressure.

At this increase in pressure the vapour is partly condensed and the gases are again absorbed by the liquid, thereby leading to the formation of zones with a high vacuum and rapid collapse of the cavities or bubbles. The surrounding liquid, hastening to fill the vacuous spaces, travels at extremely high velocities and develops a considerable dynamic head. The bubbles remaining in the cavity are not capable of stopping the motion of the particles of liquid, and the final result of this motion are blows on the walls of the channels.

The impacts are accompanied by a loss in velocity and the kinetic energy of the particle is transformed into elastic strain energy.

Owing to the negligible deformation of a liquid from an impact (for example, the module of elasticity of water is $E \approx 1 \times 10^4$ kg per sq cm) the pressure resulting from the impact may reach dozens of atmospheres.

These instantaneous processes described above may occur both in the piping and in the passageways of the pump; they are known as cavitation.

Regardless of the finish provided on the walls of the passageways in the pump and piping, cavitation will cover the surfaces with microscopic depressions called erosion. The concentrated and rapidly repeated water hammer produces minute points of mechanical destruction on the affected surfaces.

A further consequence of this mechanical destruction is increased corrosion due to the action of gases liberated from the liquid. As a matter of fact, if the corrosion resistance of a surface, provided by a protective film of oxides or other chemical compounds, is impaired, systematic water hammer will obviously increase the intensity with which the metal surface is corroded.

The result of cavitation is rapid destruction of the walls and surfaces of parts located in the zone of its action. Cavitation is commonly accompanied by a noise (roaring or hissing) and increased vibration of the pump installation, which often impairs the alignment of the pump.

With a reduction in pressure in the suction line, separate cavities, or bubbles, increase in size, merge, extend deep into the stream

of liquid and pass over into an advanced stage. Such developed cavitation leads to a decrease in output, head and efficiency of the pump.

The merging of cavities due to intensive vapour generation may result in separation of the flow of liquid, or general cavitation, after which liquid no longer enters the pump impeller and the pump ceases to operate.

General cavitation begins when p_s becomes equal to the pressure p_v of the vapours evolved from the liquid at the given temperature. This condition corresponds to the maximum suction lift $z_{smax} = z_s + h_e$ in the suction pipe and equation (182) for the case being considered can be written as

$$\frac{p_v}{\gamma} = \frac{p_a}{\gamma} - \left(z_{smax} + \frac{c_e^2}{2g} + h_{lh}\right) \tag{183}$$

Therefore

$$z_{smax} = \frac{p_a}{\gamma} - \left(\frac{p_v}{\gamma} + \frac{c_e^2}{2g} + h_{lh}\right) \tag{184}$$

The value thus obtained is called the maximum theoretical geodetic suction lift and determines the height of the impeller axis above the level of the liquid being pumped. With an increase in the temperature of the liquid the theoretically possible suction lift is reduced and, at temperatures near 100° C and with atmospheric pressure on the free surface of the liquid being pumped (entering the suction connection), the suction lift equals zero.

At high temperatures of the liquid being handled the pump impeller should be located below the liquid level so as to provide a negative lift. Practically, the permissible geodetic suction lift is the height z_s at which no cavitation phenomena are yet observed in the passages of the pump, cavitation beginning as soon as this height is exceeded.

The experience of pump manufacturing plants shows that cavitation may begin at pressures of the liquid higher than the pressure of its saturated vapour. It is therefore advisable to operate a pump with a permissible geodetic suction lift, determining the distance from the impeller axis to the free surface of the liquid, which is

$$z_{sp} = 0.75 z_{smax} \quad \text{m} \tag{185}$$

Denoting the entering velocity of the liquid at the inlet, or entry, edges of the impeller vanes as c_e we obtain

$$z_{sp} = 0.75 \left[\frac{p_s}{\gamma} - \left(\frac{p_v}{\gamma} + \frac{c_e^2}{2g} + h_{lh}\right)\right] \text{ mH}_2\text{O} \tag{186}$$

Therefore, the permissible suction lift decreases: (1) with an increase in temperature of the liquid being pumped; (2) with an increase in loss of head in the suction line; (3) with an increase in the velocity with which the liquid approaches the inlet edge of the impeller vanes; and (4) with an increase in the distance between the inlet edge and the impeller axis.

Cavitation is also promoted by sharp edges, rough passage walls, abrupt turns and other obstacles to liquid flow which force the stream away from the walls.

According to data compiled by the Leningrad Machinery Works, manometric suction lifts

$$h_{vac} = \frac{p_a}{\gamma} - \frac{p_s}{\gamma} \quad \text{mH}_2\text{O} \tag{187}$$

are listed in Table 27 for water temperatures $t < 70°$ C.

Table 27

t°C	10	20	30	40	50	60	70
h_{vac}, mH$_2$O	6	5.5	4.7	3.7	3.0	1	0

The minimum permissible surplus suction pressure Δp_s, defined as the difference between the suction pressure and the pressure of saturated vapour in the suction line of a centrifugal pump handling water at a temperature of $t > 70°$ C, may range from 0.6 to 1 kg per sq cm. The minimum surplus suction pressures are listed in Table 28.

Table 28

t°C	80	90	100	110	120	140	160	180
Δp_s, kg/cm^2	0.1	0.34	0.67	1.06	1.69	3.38	6.1	10.2

In determining the loss of head, h_l, in the suction line, the dynamic loss of head, Δh_e, at the inlet to the impeller should be taken into consideration. According to S. Rudnev

$$\Delta h_e = 10 \left(\frac{n\sqrt{Q_s}}{c_{cs}}\right)^{\frac{4}{3}} \quad \text{mH}_2\text{O} \tag{188}$$

where n = rotational speed of the pump shaft, rpm

Q_s = discharge through the impeller, cu m per sec (for double admission impellers the discharge is equal to $\frac{Q_s}{2}$)

c_{cs} = cavitation specific velocity depending upon the rotational speed of the pump and the degree of reduction of clear passage as the liquid enters the impeller. This value can be taken from Table 29.

Table 29

n_{s1}	50 to 70	70 to 80	80 to 150	150 to 200
c_{cs}	600 to 750	800	800 to 1,000	1,000 to 1,200

According to data compiled by the U.S.S.R. Institute for Hydraulic Machinery

$$\Delta h_e = \varphi_{se} \frac{c_e^2}{2g} + \Psi_{w1} \frac{w_1^2}{2g} \quad \text{mH}_2\text{O} \tag{189}$$

where $\varphi_{se} \approx 1$ to 1.2 = factor characterizing the increase in velocity at the inlet to the impeller over the mean value

c_e = mean absolute velocity at the impeller inlet

$\psi_{w1} \approx 0.3$ to 0.4 = factor characterizing the increase in the mean relative velocity w_1 at the impeller inlet

w_1 = mean relative velocity of liquid flow at the impeller inlet.

If the pump operates with negative lift, the suction pressure is increased by the pressure produced by a column of liquid of a height equal to the distance from the free surface of the liquid being pumped to the cross section of the pipeline being considered.

Normal operation of a properly designed and installed pump is substantially promoted by the absence of air inflow through leaks in the suction line, as well as the absence of accumulation of air in the pump passages and in the piping.

The parts of pumps that come into contact with cavitation zones (usually the impellers) are made of bronze or chromium stainless steel which is less susceptible to cavitation erosion than cast iron or carbon steels.

The cavitation-proof properties of a pump can be improved by:

(1) streamlining all the parts of the inlet chamber, avoiding sharp turns and projecting elements;

(2) ensuring smooth flow of the liquid over the inlet edges of the impeller vanes by proper streamlining;

(3) eliminating vortex formation, separation in flow and the formation of cavities in the liquid as it leaves the impeller vanes by streamlining the outlet edges of the vanes;

(4) reducing the relative inlet velocity w_1 to the vanes by increasing the inlet area between the vanes and by properly guiding the inlet velocities, installing inlet guide vanes at the impeller inlet (this increases c_{cs} to 1,200-1,500).

In the operation of a pumping installation, cavitation may be promoted by factors which reduce the suction pressure. These include:

(1) closed gate valves and other valves in the suction line;

(2) clogged strainers and filters;

(3) air inflow through leakages;

(4) running the impeller at a higher speed than the designed value;

(5) pressure drops in the suction line.

1-22. Centrifugal Pump Construction

A self-priming single-stage centrifugal pump combined with a water-ring vacuum pump is illustrated in Fig. 106.

The water-ring pump consists of the rotor *1*, keyed to the shaft between the removable disks *3* and *4*, and the casing *12*.

The chamber between the disks *3* and *4* communicates with cavity *11* through port *10* in disk *4*, and with the inlet connection *13* through a copper tube. The chamber also communicates with cavity *15* through port *14* in disk *3*. Cavity *15* is connected to a three-way valve which is screwed into the threaded hole of connection *17*.

The sealing device of the vacuum pump consists of the stuffing box *7*, whose compression member is the disk *4*, and rubber rings *5* and *6* fitted into recesses of disks *3* and *4*.

When the centrifugal pump is started, air is drawn from the suction line through inlet connection *13*, the copper tube, cavity *11*, and the crescent-shaped port *10* into the admission chamber of the vacuum pump. The air entering the cells between the rotor blades is compressed as it is carried over from the admission to the discharge chamber and is forced through port *14* into cavity *15*. From there it passes through connection *17*, the three-way valve and the venting pipe into the atmosphere.

One of the openings of the three-way valve is connected, as described above, to the air pipe and the other to a pipe leading to hole *2* of the inlet connection *13*.

As soon as the liquid being pumped fills (primes) the admission cavity of the centrifugal pump it begins to discharge from the venting pipe. When the centrifugal pump develops the required capacity, the three-way valve is switched over to connect the discharge of the vacuum pump with the inlet connection *13*. This is done to avoid loss of the liquid through the vacuum pump. During operation of the centrifugal pump liquid must circulate through the vacuum pump to prevent it from overheating. Passage *16* in disk *3*, connecting the working chamber of the vacuum pump with cavity *15*, enables a constant liquid level to be maintained in this pump so that the rotor blades are always properly immersed in the liquid.

Passages *18*, *19* and *9*, and plug *20* serve to drain the liquid from the casing of the vacuum pump whenever necessary.

If the vacuum pump shown in Fig. 106 is removed, we obtain an ordinary nonpriming single-stage centrifugal pump.

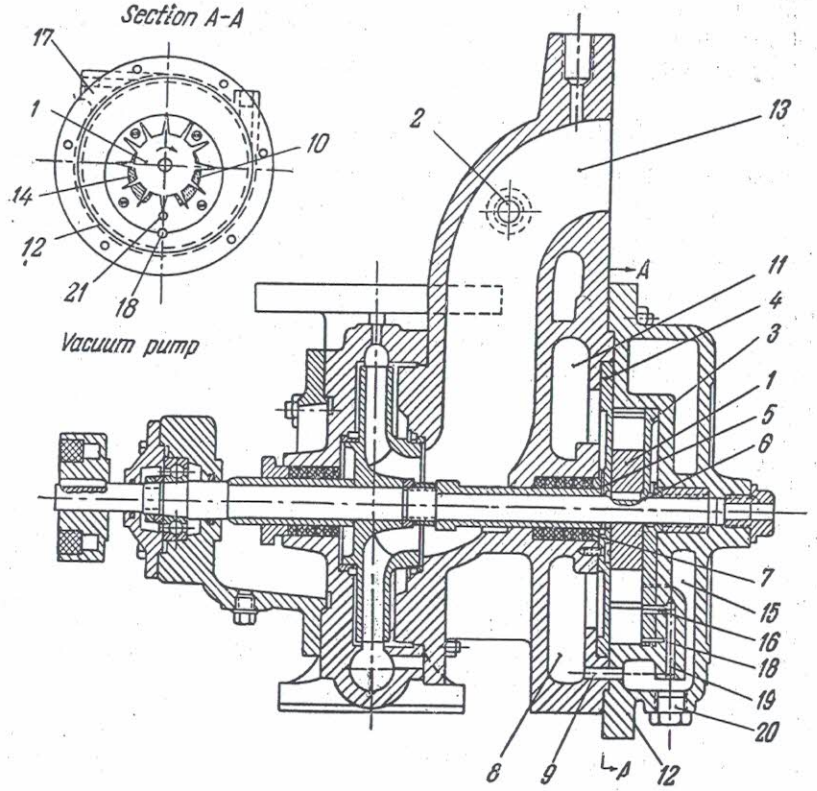


Fig. 106.

As mentioned previously, several impellers of a pump may be arranged to operate in series to generate the required head.

It is not always possible to reduce the discharge velocity of high-speed and high-pressure pumps to the required velocity of flow in the discharge pipe by the use of a volute casing. For this purpose it is sometimes necessary to use a recuperating device called a diffuser-, or guide-ring, through which the liquid leaving the impeller passes before it enters the volute. Such a diffuser-ring is shown schematically in Fig. 107.

From the vanes *1* of the impeller the liquid enters the passages *2* between the guide vanes *3* of the diffuser-ring.

As mentioned above, the output of a centrifugal pump can be increased by arranging the impellers for parallel operation, and the end thrust can be balanced by double admission to the impeller. A pump designed along these lines, with double admission and with

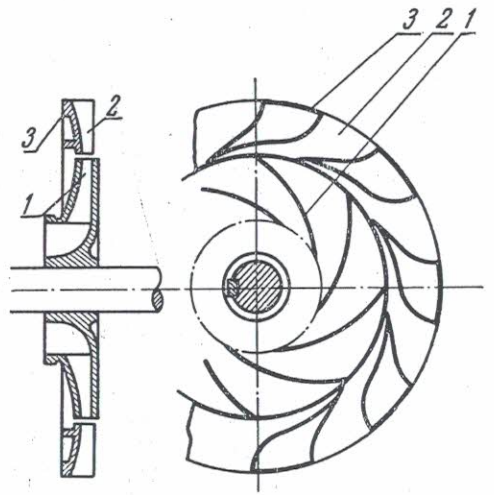


Fig. 107.

a diffuser-ring and volute casing, is illustrated in Fig. 108. Another feature of this pump is the sectional casing with removable side covers.

The casing comprises the inlet connection *1*, cast integral with the volute casing *2* and the side covers *5* and *6*. These parts are assembled into a single unit and secured by studs *3* and nuts *4*. Bearing housings *7* and *8* are secured to the side walls *5* and *6* by studs *9* and nuts *10*. The double admission pump impeller *12* is mounted on shaft *11* and fixed by key *13*. The impeller is clamped on the shaft between shoulder *14* and the shaped nut *15*, screwed on the threaded portion of the shaft. The shaft runs in split plain bearings *16* which have babbitted liners *17*. Lubricant is supplied to the bearings by oil rings *18* from the lower cavities of the bearing housings which serve as oil baths. The latter are filled through the covers *19*.

Rings *21*, secured to the pump shaft by screws *22*, seal the opening where the shaft passes out of the bearing housings.

Oil leaking from the housings is collected in pans and drained through threaded plugs *23*. The housings can be drained through

holes closed by plugs *24*. The pump shaft is sealed by stuffing boxes in which the packing *31* is divided into two parts by the lantern rings *25* to which liquid is supplied by tubing *26* to seal the packing. Bronze bushings *28* are fitted into the glands *27*, and bronze throat bushings are provided where the shaft passes through the pump casing. The packing is compressed by the glands *27* when nuts *29* are tightened on studs *30*.

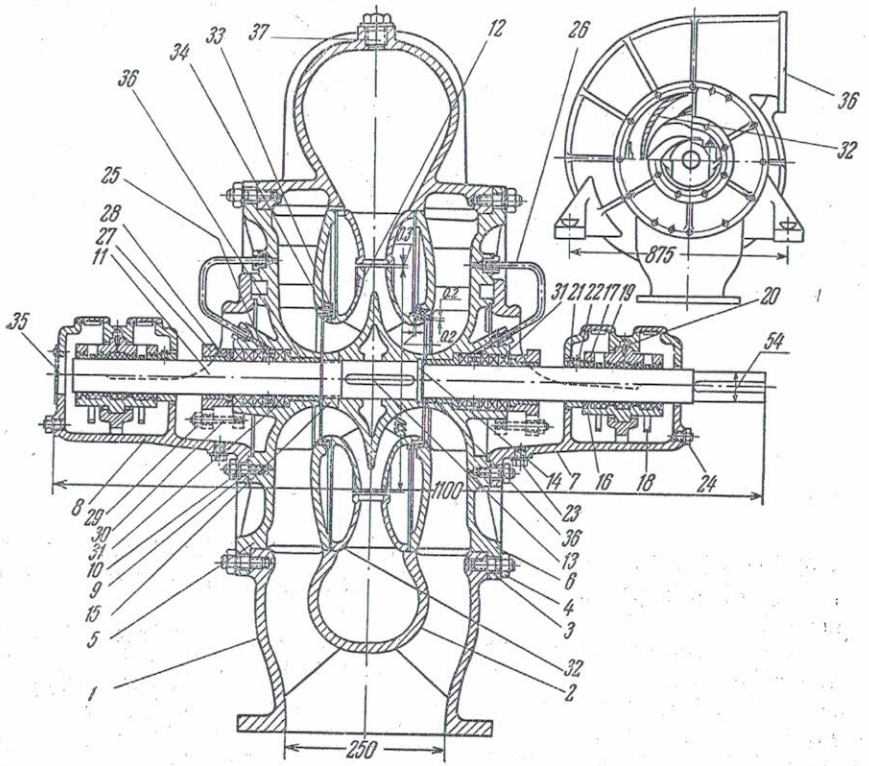


Fig. 108.

Bronze wearing rings *33*, fastened by screws *34*, are fitted in the casing to avoid damage to the impeller if it shifts to one side, to maintain the proper clearance and to restore this clearance during repairs.

Upon leaving the impeller the liquid first passes through the diffuser-ring *32* and then through the volute casing *2* into the discharge connection *36* and the delivery line.

The volute has a plug *37* for priming the pump and bleeding the air from the casing. A test cock with a funnel can be installed in place of plug *37*. Access to the pump shaft to measure its rotational speed is possible through cover *35*.

Horizontal centrifugal pumps are extensively employed because they are more convenient in assembly and disassembly.

It has been previously noted that the head of a pump is increased by arranging its impellers for series operation.

A horizontal two-stage centrifugal pump is shown in Fig. 109. Two impellers, *6* and *13*, are mounted on the shaft *21* and housed within the pump casing.

The pump casing comprises the cover *3*, cylindrical section *4* and cover *5* held together by tie bolts *1* with nuts *2*.

Cover *3* serves as the casing for the first-stage impeller *6* and its diffuser-ring *7* for converting dynamic into static head. The casing of the inlet connection *8* is secured to cover *3* while the housing of the ball thrust bearing *9* and radial bearing *10* is fastened to casing *8*. The cylindrical section *4* contains the passages with the inlet guide vanes *12*, connecting the discharge chamber *15* of the first stage to the inlet chamber *16* of the second-stage impeller *13*, the impeller itself and its diffuser-ring *14*. Cover *5*, cast integral with the discharge connection *11* and collecting chamber *17*, is secured to the cylindrical section *4* of the casing. Housing *18* of the plain bearing with ring lubrication is mounted on the cover *5*. The stuffing boxes are arranged in the covers. The pump impellers have holes *19* for equalizing the pressure on both sides of the impeller disks, thereby eliminating unbalanced end thrust. The left stuffing box has a hydraulic seal ensuring proper tightness and avoiding air inflow. Tube *20* supplies liquid to the hydraulic seal.

The main feature of the design of the casing in this pump is the ease with which worn parts of the casing can be replaced.

In pump operation the liquid entering the inlet connection *8* passes through the stationary guide vanes *22* to the vanes of first-stage impeller *6* where it acquires a part of the mechanical energy applied to the pump shaft by the motive unit. Leaving the vanes of impeller *6*, the liquid enters diffuser-ring *23*, consisting of separate stationary vanes. As the liquid flows through the diffuser-ring its diverging passages reduce the flow velocity of the liquid, thereby converting part of the kinetic energy of the liquid into potential energy. Thus, the dynamic head of the liquid is reduced and the static head increases.

From diffuser-ring *23* the liquid flows through circumferential chamber *15* and enters the stationary return passages with vanes *12* which guide the liquid and impart a definite direction to its velocity. Next, the liquid enters the second-stage impeller *13* in which the

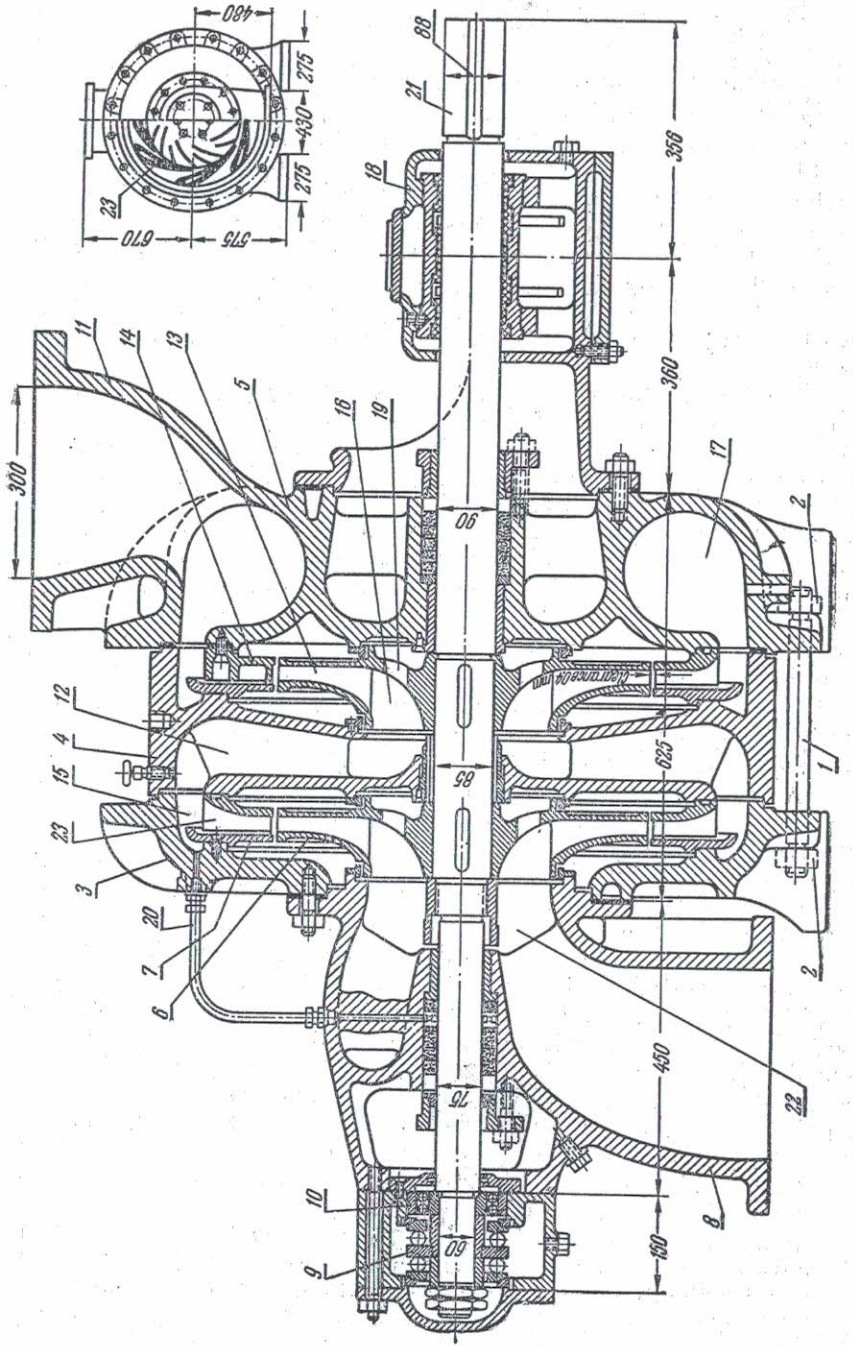


Fig. 109.

energy is again increased as it was in the first stage. Diffuser-ring *14* again converts a part of the kinetic energy into potential energy, or dynamic head into static head. Chamber *17* in cover *5* collects the liquid leaving the diffuser-ring vanes and from here it is discharged through connection *11* to the point of delivery. Thus, each impeller develops half of the total head of the pump.

The head developed by the pump can be conveniently increased by adding identical sections operating in series.

A sectional, or ring-type, four-stage centrifugal pump is shown in Fig. 110. Its casing comprises cover *5*, cylindrical ring-type sections *6* and cover *14* held together by studs *8* and nuts *4*. The distinguishing feature of this pump is the sectional casing in which each section *6* is a single-stage centrifugal pump.

The convenience offered by such a pump in maintenance is that any section that has got out of order can be readily replaced with a spare section. Sections can be easily added whenever necessary to increase the head developed by the pump.

The pump casing houses the impellers *3* which are fixed on shaft *17* by keys. Shaft *17* runs in plain bearings *18* and *20*. End thrust that is not balanced by the provision of holes *15* in the impeller disks is carried by the ball thrust bearing *1*.

Bearing *18* is lubricated by oil rings *16*.

Each section of the pump has a diffuser-ring *9* to convert dynamic head into static head (velocity energy into pressure energy) and return passages *10* to guide the liquid flow to the next stage. The liquid being handled enters inlet connection *19* and passes consecutively through the inlet chamber *2*, the passages of impeller *3* and diffuser-ring *9*, return passages *10* and circumferential chamber *11*, from where it is discharged through connection *12*. Air is bled from the pump and its operation is checked by connecting test cocks to channels *7* and a pressure gauge to channel *13*.

It was previously pointed out that a centrifugal pump has a high suction lift in steady operation but cannot operate with dry suction. The efficiency of a centrifugal pump is from 2 to 2.5 times higher than that of a turbine pump. On the other hand, a turbine pump can operate with dry suction and can develop a head from 3 to 5 times higher than that of a centrifugal pump operating at the same rotational speeds.

The suction lift of a turbine pump is limited to 4-6 mH_2O in steady operation and is lowered due to the high head at the inlet to the impeller as a result of the restricting counter and transverse vortex currents in the liquid at the inlet regions of the impeller.

A combination centrifugal-turbine pump enables the advantages of both types of pumps to be utilized. Such a pump is illustrated in Fig. 111.

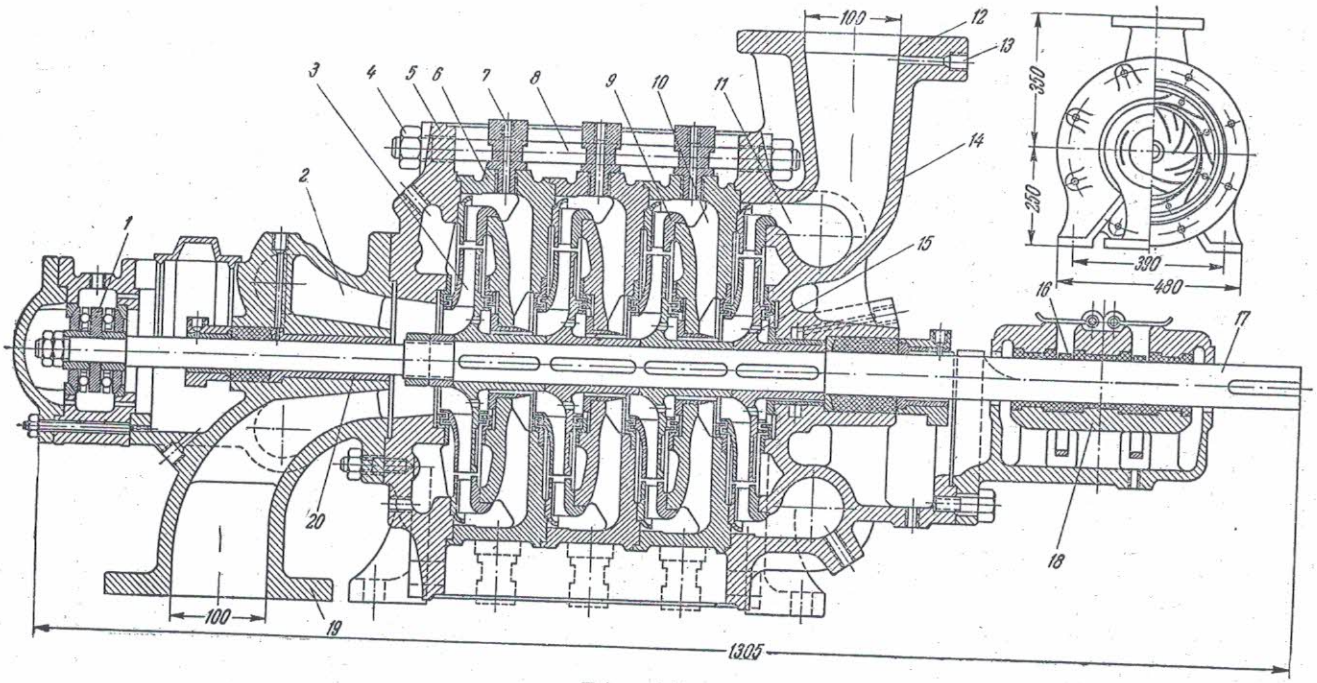


Fig. 110.

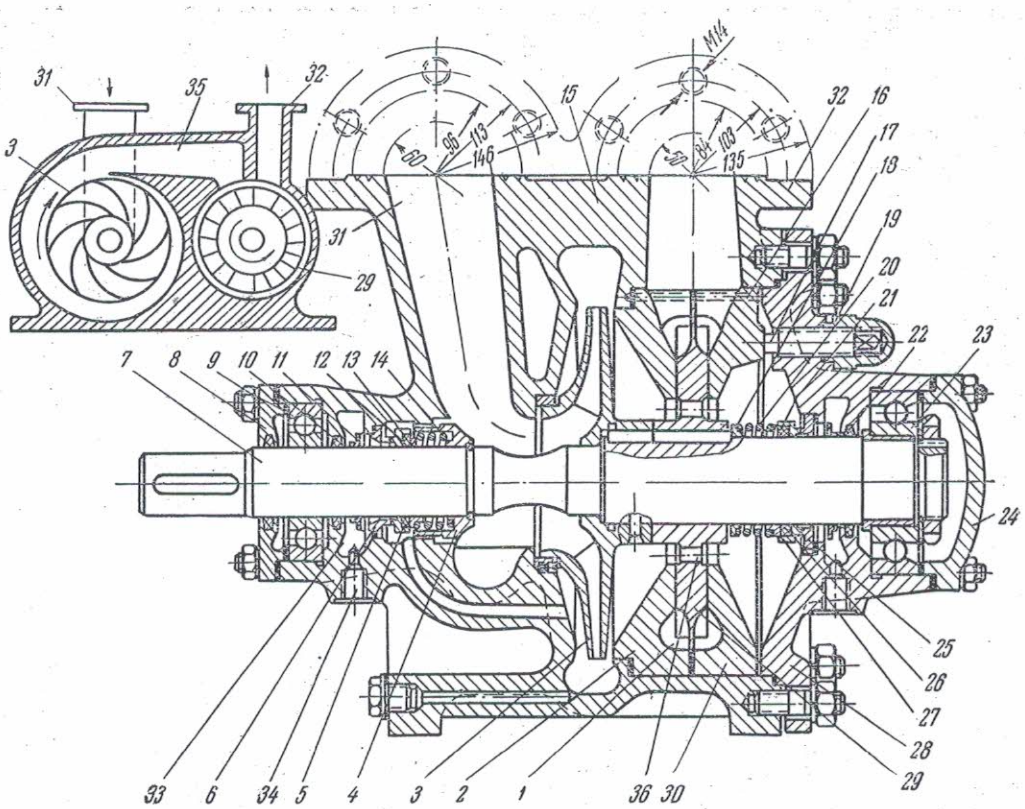


Fig. 111.

The casing *15* of the pump houses the centrifugal impeller *3* and the turbine impeller *29* with intervane spaces, or cells, on both sides, mounted on the same pump shaft *7*. When the pump is started, impeller *29* draws air from the suction line through connection *31*, impeller *3* and passage *35* to create a vacuum in the suction line.* This vacuum ensures liquid flow to the centrifugal impeller which discharges through passage *35* to the turbine impeller. The latter, in turn, discharges the liquid along annular channel *1* and through connection *32* into the discharge line. The outlet channel *1* of impeller *29* is formed by inserts *2* and *30* fixed by dowel pin *16* and three screws *17*, which are enclosed on the outside by cap nuts *19*. This arrangement provides for maximum convenience in assembly, repair and installation of impellers and inserts. Shaft *7* runs in the radial ball bearing *9* and angular-contact ball bearing *23* fixed by covers *8* and *24*, respectively.

Felt seals *10* and *22* retain grease in the bearings. Fine-felt seals with higher wear resistance may also be used. The pump shaft has two mechanical seals.

The seal of the centrifugal pump rotates together with the shaft and consists of the bronze retaining sleeve *4*, brass guide ring *13*, and rubber sealing ring *5* inserted in bronze thrust ring *12*. Helical spring *14* in sleeve *4* holds thrust ring *12*, together with its components, against the stationary seat member *6*. Liquid leaking past sleeve *4* is thrown by slinger *11* into chamber *33* from where it is drained through hole *34*. The mechanical seal of the turbine pump consists of rings *18* and *21*, rotating together with shaft *7*, rubber sealing ring *26* inserted in thrust ring *27*, and helical spring *20*, holding thrust ring *27* against the stationary seat member *25*. Liquid that leaks past the seal is thrown by slinger *11* mounted on shaft *7*. Through-channel *36* balances the end thrust on impeller *29*. Ball bearing *23* is mounted in cover *28* and is fixed axially by cover *24*. Combination centrifugal-turbine pumps are available for capacities from 0.4 to 34 cu m per hour and operate against heads from 53 to 224 mH_2O (less frequently, up to 500 mH_2O).

1-23. Head Developed by a Centrifugal Pump Impeller

We shall use Fig. 112 to determine the theoretical head developed by a centrifugal pump.

The liquid being pumped passes from the central chamber *1* in the pump casing to the stationary inlet guide vanes *2*. In the passages between these vanes, the dynamic head of individual streams in the

* For the sake of clarity both impellers of the pump are shown in a single plane in the diagram of this pump.

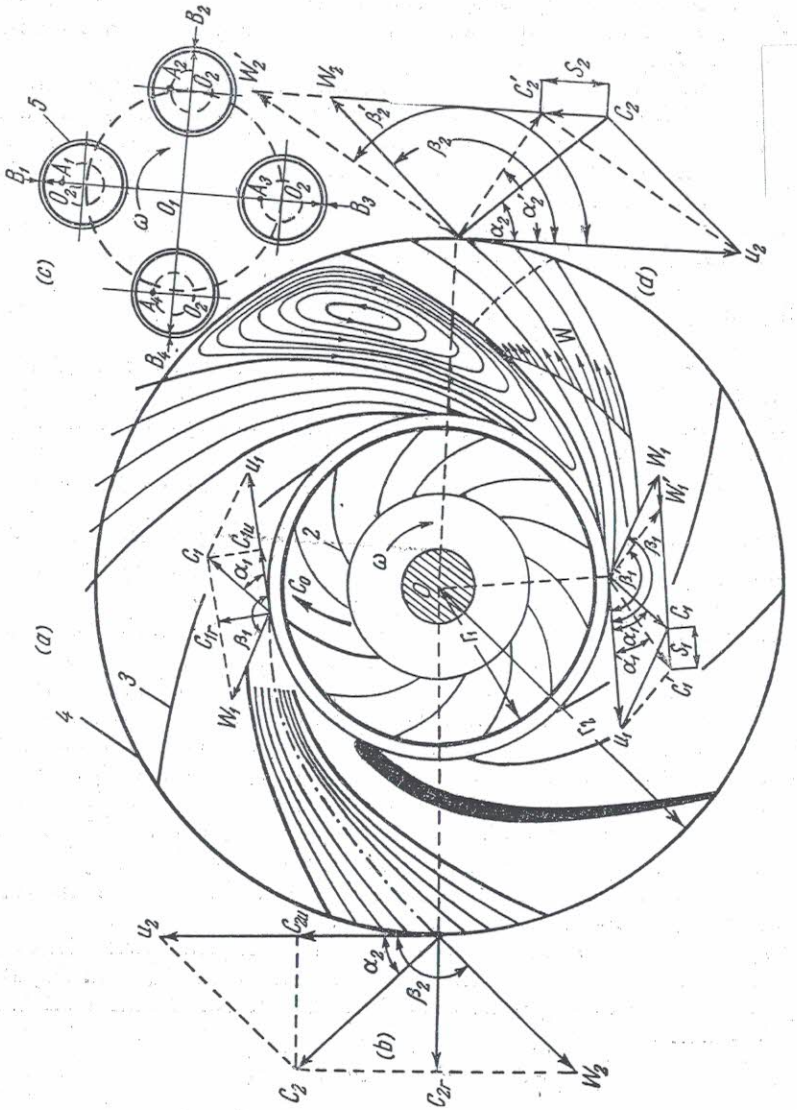


Fig. 112.

liquid flow is partly converted into static head and the velocity of the particles of the liquid is given a definite direction so that the particles, or elements, slide smoothly onto the vane *3* of rotating impeller *4* without shock.

Let us assume that impeller *4* has an infinite number of infinitely thin vanes *3* of identical shape (Fig. 112*a*). In this case the liquid will be divided into an infinitely large number of streams as it enters the impeller. Elements of the liquid, flowing through the intervane passages from the inlet circumference of radius r_1 to the outlet circumference of radius r_2, will describe identical paths. The liquid in the intervane passages will participate in transfer, relative and absolute motion due to the mechanical action exerted on the elements of liquid by the vanes of the impeller as it rotates about the axis *O* with an angular velocity ω.

In entering onto the vane and leaving the vane each element of liquid (Fig. 112*b*) acquires the following:

(1) peripheral velocities u_1 and u_2 tangent to the inlet and outlet circumferences of the vanes, respectively;

(2) relative velocities w_1 and w_2 tangent to the profile of the vane surface;

(3) absolute velocities c_1 and c_2 obtained as the geometric sums of u_1 and w_1 and u_2 and w_2, respectively, and directed at angles α_1° and α_2° to the respective peripheral velocities.

The energy (head) imparted to the liquid as it flows through the intervane passages can be expressed as the difference in the moments of momentum at the inlet and outlet circumferences of the vanes. According to the relevant law of mechanics, the increment of the moment of momentum of a material system, in reference to a given axis in a certain interval of time, is equal to the moment of impulse, relative to the same axis, of all external forces acting on the system.

Let us denote the angles between the peripheral and relative velocities at the inlet and outlet circumferences as β_1° and β_2°, respectively. Now we can construct the diagram of velocities of the elements of liquid in the intervane passages.

If we denote by *m* the mass of the inviscid liquid flowing through the impeller in one second, then the moments of momentum M_1 and M_2 in reference to the shaft axis *O* at the inlet and outlet circumferences will be

$$M_1 = r_1 m c_{1u} = r_1 m c_1 \cos\alpha_1^\circ$$

and

$$M_2 = r_2 m c_{2u} = r_2 m c_2 \cos\alpha_2^\circ$$

Any increase in the moment of momentum of the liquid stream in the pump impeller can only be due to the mechanical energy

developed by the impeller drive and imparted to the liquid. The increment of the moment of momentum is

$$\Delta M = M_2 - M_1 = r_2 m c_2 \cos\alpha_2^\circ - r_1 m c_1 \cos\alpha_1^\circ$$

If the power *N* kg-m per sec, applied to the impeller shaft, was transmitted fully to the liquid, the torque on the pump shaft would be

$$M = \frac{N}{\omega} \text{ kg-m}$$

The moment of impulse of the external forces during the time $t=1$ sec is $Mt=M$. Therefore, according to the law of moments of momentum

$$M = \frac{N}{\omega} = r_2 m c_2 \cos\alpha_2^\circ - r_1 m c_1 \cos\alpha_1^\circ \qquad (190)$$

The theoretical head, $H_{t\infty}$, generated by the pump impeller is defined as the difference in the heads of the liquid at the outlet and inlet circumferences of the impeller.

To distinguish it from the actual head, *H*, developed by the impeller the theoretical head has the subscripts *t* and ∞. The subscript *t* indicates that the theoretical head does not take into consideration the loss in head when the liquid flows through the impeller passages (the assumption of an inviscid liquid) while the subscript ∞ indicates that the impeller has an infinite number of vanes.

Since all the power applied to the impeller is transmitted to the liquid in the intervane passages, we can write

$$N = mgH_{t\infty} \qquad (191)$$

Combining equations (190) and (191), after multiplying both sides of equation (190) by ω, we obtain

$$\frac{mgH_{t\infty}\omega}{\omega} = \omega r_2 m c_2 \cos\alpha_2^\circ - \omega r_1 m c_1 \cos\alpha_1^\circ \qquad (192)$$

whence

$$H_{t\infty} = \frac{u_2 c_2 \cos\alpha_2^\circ - u_1 c_1 \cos\alpha_1^\circ}{g} = \frac{u_2 c_{2u} - u_1 c_{1u}}{g} \qquad (193)$$

This equation was deduced by Leonhard Euler in 1754 and is the basis for the jet theory of turbines. Various problems concerned with centrifugal pump operation, calculation and design can be solved by simple methods on the basis of this theory. As is well known, the head of a stream of liquid consists of static and dynamic heads. The principal energy of the liquid, lost in overcoming hydraulic resistances in the system through which it flows, is expressed by the static head while the rate of flow is expressed by the dynamic head.

To solve various problems of pump operation it is necessary to know, not only the total head $H_{t\infty}$, but its dynamic H_{dyn} and static H_{st} components as well.

We shall now find the expressions for $H_{t\infty st}$ and $H_{t\infty dyn}$. Using the velocity diagrams at the inlet and outlet circumferences of the impeller vanes we obtain

$$w_1^2 = c_1^2 + u_1^2 - 2u_1c_1\cos\alpha_1^\circ$$

whence

$$u_1c_1\cos\alpha_1^\circ = \frac{u_1^2 - w_1^2 + c_1^2}{2} \qquad (194)$$

and

$$w_2^2 = c_2^2 + u_2^2 - 2u_2c_2\cos\alpha_2^\circ$$

whence

$$u_2c_2\cos\alpha_2^\circ = \frac{u_2^2 - w_2^2 + c_2^2}{2} \qquad (195)$$

Combining equations (193), (194) and (195), we can write

$$H_{t\infty} = \frac{u_2c_2\cos\alpha_2^\circ - u_1c_1\cos\alpha_1^\circ}{g} = \frac{u_2^2 - u_1^2}{2g} + \frac{w_1^2 - w_2^2}{2g} + \frac{c_2^2 - c_1^2}{2g} \qquad (196)$$

where $\frac{u_2^2 - u_1^2}{2g}$ = increment of the static head due to the inertia forces appearing when the elements of liquid pass from one radius of the impeller to the next. This head is always positive in value

$\frac{w_1^2 - w_2^2}{2g}$ = increment of the static head due to the change in the velocity of the liquid in the intervane passage which results, in turn, from the change in clear cross-sectional area of the passage. This term may be either positive or negative in value

$\frac{c_2^2 - c_1^2}{2g}$ = increment in dynamic head due to the change in absolute velocity of liquid flow in the intervane passage of the impeller. This head is positive in value.

The approximate ratios between $H_{t\infty st}$, $H_{t\infty dyn}$ and $H_{t\infty}$ for available pumps are

$$H_{t\infty st} = (0.7 \text{ to } 0.8)\,H_{t\infty} \qquad (197)$$

$$H_{t\infty dyn} = (0.2 \text{ to } 0.3)\,H_{t\infty} \qquad (198)$$

$$\frac{H_{t\infty st}}{H_{t\infty dyn}} = 3.3 \text{ to } 4 \qquad (199)$$

Euler's equation does not take into consideration the effect of the finite number of vanes and the loss of head in the impeller on the head developed by the impeller. Because the number of vanes is finite, the path of an element of liquid in the passage of the rotating impeller, far from the outline of the vane, will not conform to the shape of the vane. Moreover, a rotary motion of the liquid is observed in the impeller passages. We shall now refer to Fig. 112*c* to explain the origin of this motion.

Let us assume that a vessel 5 is filled with liquid and is rotated with a constant angular velocity of ω about point O_1. In this case, each element of liquid in the vessel travels in its corresponding circle, concentric with the walls of the vessel, in the opposite direction but at the same velocity. When the vessel makes one full revolution about the centre O_1 each element makes one full revolution about the centre O_2 of the vessel. In fact, if a point B_1 is marked on the wall of the vessel opposite element A_1, then B_1 will consecutively occupy the positions B_2, B_3 and B_4 upon rotation of the vessel. It is evident that element A_1 of the liquid has a rotary motion in reference to the surface of the vessel, consecutively occupying the positions A_1, A_2, A_3, and A_4.

A similar movement of the elements of the liquid in the reverse direction to impeller rotation takes place in the passages of an impeller rotating with an angular velocity ω about the centre O. This motion is called counter rotation, or relative whirl. If the impeller rotates clockwise, the rotary motion of the liquid will be counterclockwise and at the same angular velocity (Fig. 112*c*).

In this manner, the value of the relative velocity, w_i, at the leading surface of the vane in the direction of rotation is reduced while that on the opposite (rear) surface of the vane is increased as is clearly shown in Fig. 112*d*.

Therefore, on the basis of Bernoulli's equation, it can be contended that the pressure on the leading surface of the vane is higher than on the opposite surface.

Counter rotation will contribute a certain velocity S_1 in the direction of the peripheral velocity u_1 to the absolute velocity c_1 at the inlet circumference of the impeller, and a certain velocity S_2 in the direction opposite to the peripheral velocity u_2, at the outlet circumference.

Denoting the changed absolute velocities as c_1' and c_2' and their projections in the direction of the peripheral velocities as c_{1u}' and c_{2u}', respectively, we obtain a velocity diagram for a finite number of vanes (Fig. 112*d*). Here, the head developed by the pump impeller is

$$H_t = \frac{u_2c_2'\cos\alpha'^\circ_2 - u_1c_1'\cos\alpha'^\circ_1}{g} = \frac{u_2c_{2u}' - u_1c_{1u}'}{g}$$

According to A. Stodola

$$S_1 = \varepsilon_{cr} \frac{\pi u_1}{z_v} \sin \beta_1^\circ$$

and

$$S_2 = \varepsilon_{cr} \frac{\pi u_2}{z_v} \sin \beta_2^\circ$$

where z_v = number of vanes in the impeller
$\varepsilon_{cr} \approx 0.6$ to 0.8 for low-pressure pumps
$\varepsilon_{cr} \approx 0.7$ to 1.2 for high-pressure pumps.
The angles of the relative velocity at inlet and outlet are

$$\beta_1'^\circ = \beta_1^\circ - (1^\circ \text{ to } 3^\circ)$$
$$\beta_2'^\circ = \beta_2^\circ + (10^\circ \text{ to } 15^\circ)$$

In practice the effect of the finite number of vanes is taken into account by the circulation factor, k_{2cu}.

Assuming that the difference in pressure on the two sides of the vane is constant along the entire length of the vane, Prof. C. Pfleiderer has found the following value of this factor:

$$k_{2cu} = \frac{1}{1 + 2\frac{\varphi_u}{z_v} \frac{1}{1 - \left(\frac{r_1}{r_2}\right)^2}}$$

The circulation factor, according to data compiled by Academician G. Proskura, is

$$k_{2cu} = 2\frac{\varphi_u}{z_v} \frac{1}{1 - \left(\frac{r_1}{r_2}\right)^2}$$

where φ_u = (0.55 to 0.65)+0.6 sin $(180^\circ - \beta_2^\circ)$=a correction factor in which the first term depends on the surface finish of the pump passages. The higher the class of finish, the lower the value of the factor will be.

A reduction in the head developed by the pump due to the loss in head in the pump passages is taken into consideration by the hydraulic efficiency, η_h. Thus the actual head generated by the pump impeller is

$$H = k_{2cu}\eta_h H_{t\infty} = k_{2cu}\eta_h \frac{u_2 c_2 \cos\alpha_2^\circ - u_1 c_1 \cos\alpha_1^\circ}{g} = k_{2cu}\eta_h \frac{u_2 c_{2u} - u_1 c_{1u}}{g} \quad (200)$$

Data, resulting from investigations made by N. Zhumakhov concerning the variation in η_h and k_{2cu} with the specific velocity n_{s1} are listed in Table 30.

Table 30

n_{s1}	60 to 100	100 to 150	150 to 220
η_h	0.87 to 0.9	0.9 to 0.92	0.92 to 0.94
k_{2cu}	0.60 to 0.82	0.82	0.82

Most centrifugal pumps do not have inlet guide vanes. In this case, the absolute velocity of inlet onto the vanes is directed almost along a radius, i. e., $\alpha \approx 90^\circ$. As a result

$$H_{t\infty} = \frac{u_2 c_2 \cos\alpha_2^\circ - u_1 c_1 \cos\alpha_1^\circ}{g} = \frac{u_2 c_{2u}}{g} \quad (201)$$

and

$$H = k_{2cu}\eta_h \frac{u_2 c_2 \cos\alpha_2^\circ}{g} = k_{2cu}\eta_h \frac{u_2 c_{2u}}{g} \quad (202)$$

After multiplying the numerator and denominator of equation (202) by u_2 we obtain

$$H = k_{2cu}\eta_h \frac{u_2 c_{2u}}{g}\frac{u_2}{u_2} = k_{2cu}\eta_h \frac{u_2^2}{g}\frac{c_{2u}}{u_2} = k_h \frac{u_2^2}{g} \quad (203)$$

where $k_h = k_{2cu}\eta_h \frac{c_{2u}}{u_2}$ = head factor, equal to 0.4-0.55 for pumps with inlet guide vanes and 0.3 to 0.5 for pumps without such vanes.

Thus, the head developed by pumps having the same head factor, k_h, depends upon the peripheral velocity, u_2, at the outlet circumference of the impeller. The peripheral velocity, u_2, increases with the rotational speed, n, of the impeller and with its radius r_2.

At the present time the metallurgical industry provides materials for impellers that permit the use of maximum outlet peripheral velocities from 220 to 250 m per sec.

The maximum rotational speed of an impeller is determined by its ability to operate without cavitation at that speed. Any large increase in the radius of the impeller is undesirable because of the increased loss of head from friction of the impeller shrouds on the liquid.

An increase in the head factor is associated with an increased absolute velocity c_2 at the output from the impeller. This increases

the loss of head in the impeller passages and the losses in the diffuser-ring at the impeller outlet due to hydraulic friction and shock in converting dynamic into static head.

If the optimal values are selected for the factors upon which the head developed by a pump impeller depends, an actual head $H=300$ mH_2O and more can be obtained for a single impeller.

1-24. Effect of the Outlet Angle of the Relative Velocity on the Head Developed by the Pump

Let us see what influence variations in the outlet angle of the relative velocity have on the head generated by the pump impeller. This can be done by expressing the velocity c_{2u} in equation (201) in terms of the angle β_2° (Fig. 113*a*). On the basis of the velocity parallelogram at the outlet circumference of the impeller we can write

$$\tan\beta_2^\circ = \frac{c_{2r}}{c_{2u}-u_2}$$

or

$$(c_{2u}-u_2)\tan\beta_2^\circ = c_{2r}$$

then

$$c_{2u} = \frac{u_2\tan\beta_2^\circ + c_{2r}}{\tan\beta_2^\circ} = u_2 + c_{2r}\cot\beta_2^\circ \qquad (204)$$

Substituting equation (204) into (201) we obtain

$$H_{t\infty} = \frac{u_2c_{2u}}{g} = \frac{u_2^2}{g} + \frac{u_2c_{2r}}{g}\cot\beta_2^\circ \qquad (205)$$

Let us assume that only angle β_2° is varied on the impeller vanes and that their geometric dimensions, capacity and rotational speed remain constant. Then, using equation (205) we obtain (Fig. 113):

(a) for forward-curved vanes (Fig. 113*a*) $\beta_2^\circ < 90°$ and

$$H_{t\infty} > \frac{u_2^2}{g}$$

(b) for straight, or radial, vanes (Fig. 113*b*) $\beta_2^\circ = 90°$ and

$$H_{t\infty} = \frac{u_2^2}{g}$$

(c) for backward-curved vanes (Fig. 113*c*) $\beta_2^\circ > 90°$ and

$$H_{t\infty} < \frac{u_2^2}{g}$$

Thus, impellers with forward-curved vanes develop the highest head.

On the other hand, the power required to rotate an impeller of this type for a given rotational speed and capacity will also be higher, since $N=mgH_{t\infty}$.

To sum up, it can be stated that the head can be varied at constant geometrical dimensions and rotational speed of the impeller by using vanes with the corresponding outlet angle β_2 of the relative velocity w_2.

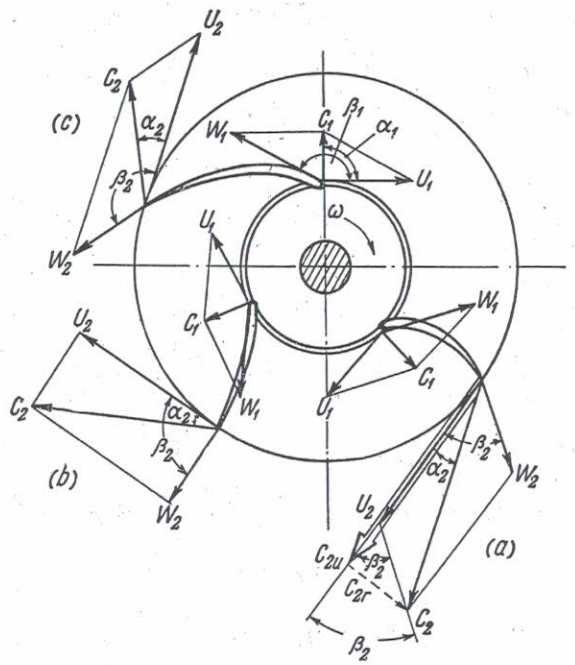


Fig. 113

Next, we shall consider the effect of angle β_2 on variations in the static and velocity heads (Fig. 114).

As a rule, the absolute inlet velocity c_1 onto the vanes is taken equal to the radial component of the absolute outlet velocity c_{2r} from the vanes.

Applying these equivalent velocities we can write the expression for the dynamic head in the following form:

$$H_{t\infty dyn} = \frac{c_2^2 - c_1^2}{2g} = \frac{c_2^2 - c_{2r}^2}{2g} \qquad (206)$$

It follows from the velocity parallelogram (Fig. 113*a*) that

$$c_2^2 - c_{2r}^2 = c_{2u}^2 = (u_2 + c_{2r}\cot\beta_2^\circ)^2 \qquad (207)$$

Combining equations (206) and (207) we obtain

$$H_{t\infty dyn} = \frac{(u_2 + c_{2r}\cot\beta_2^\circ)^2}{2g} \qquad (208)$$

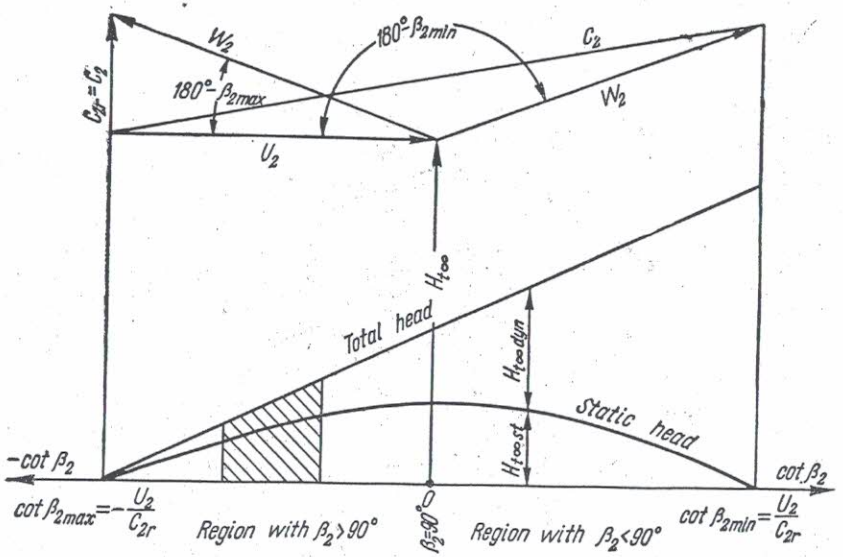


Fig. 114.

Therefore, the static head $H_{t\infty st}$ is

$$H_{t\infty st} = H_{t\infty} - H_{t\infty dyn} = \frac{u_2^2}{g} + \frac{u_2 c_{2r}\cot\beta_2^\circ}{g} - \frac{u_2^2}{2g} - \frac{2u_2 c_{2r}\cot\beta_2^\circ}{2g} - \frac{(c_{2r}\cot\beta_2^\circ)^2}{2g} = \frac{u_2^2}{2g} - \frac{(c_{2r}\cot\beta_2^\circ)^2}{2g} \qquad (209)$$

By plotting the head values along the axis of ordinates and the values of cot β_2° along the axis of abscissas we can construct the curve of the variation of the head developed by the impeller. When the projection of velocity w_2 on the peripheral velocity $c_{2r}\cot\beta_2 = w_{2u}$ becomes equal to the peripheral velocity u_2 and is oppositely directed, the pump impeller will develop neither static nor dynamic head (see equations 207 and 209).

Taking equation (205) equal to zero we obtain the angle β_{2max}° at which the impeller does not develop a head. Thus

$$0 = \frac{u_2^2}{g} - \frac{u_2 c_{2r}\cot\beta_{2max}^\circ}{g}$$

and

$$-\cot\beta_{2\,max}^\circ = \cot(180° - \beta_{2\,max}^\circ) = \frac{u_2}{c_{2r}} \qquad (210)$$

In this manner, we find the initial point of the curve.

Now we shall consider the character of head variation for a variable angle. The variation in total head upon a decrease in angle β_2° is represented, according to equation (205), by a straight line. Using equations (205), (208) and (209) we obtain the following for an angle $\beta_2^\circ = 90°$ (radial vanes):

$$H_{t\infty} = \frac{u_2^2}{2g};\quad H_{t\infty st} = \frac{u_2^2}{g} \text{ and } H_{t\infty dyn} = \frac{u_2^2}{2g}$$

Further decrease of angle β_2° will lead to an increase in the dynamic head (equation 208) and a decrease in the static head (equation 209). When the dynamic head becomes equal to the total head, the static head is reduced to zero (equation 205). This value of the angle ($\beta_{2\,min}^\circ$) can be found by taking equation (209) equal to zero:

$$\frac{u_2^2}{2g} - \frac{(c_{2r}\cot\beta_{2min}^\circ)^2}{2g} = 0 \text{ or } \cot^2\beta_{2\,min}^\circ = \frac{u_2^2}{c_{2r}^2}$$

whence

$$\cot\beta_{2\,min}^\circ = \frac{u_2}{c_{2r}} \qquad (211)$$

A still further decrease in angle β_2° leads to a separation in the stream of liquid, as is indicated by the negative value of the total head obtained in equation (209).

Thus, theoretically, the chief means for varying the head at a given rotational speed and geometric dimensions of the impeller is by varying the outlet angle β_2° which may range from 45° to 170°.

We see, therefore, that an increase in angle β_2° increases the static head and decreases the dynamic head and, consequently, the outlet velocity, c_2, thereby facilitating its reduction to the velocity in the discharge line.

On the other hand, a decrease in angle β_2° decreases the static head and increases velocity c_2. This, in turn, increases the hydraulic losses and reduces the efficiency of the pump. Moreover, difficulties

arise in the subsequent reduction of c_2 to the velocity in the discharge line.

The higher the dynamic head that must be converted into static head, the higher the losses in head will be. Besides, the use of forward-curved vanes increases the hydraulic losses in the impeller because of the sharp changes of direction in the passages and their rapid divergence.

Backward-curved vanes provide for a smoother change in the velocities of the liquid flow and their more uniform distribution in the impeller passages, thereby reducing hydraulic losses in the impeller. Pumps with backward-curved vanes are capable of quieter operation with a more stable relationship between the capacity and the head developed by the pump.

These considerations have led to the general use, at the present time, of centrifugal pumps with impellers having backward-curved vanes. The most frequently employed outlet angles β_2° range from 160° to 170°. The hatched area in Fig. 114 shows the ratio of the heads for this range of angles.

1-25. Centrifugal Impeller Vanes

Impeller vanes, bounded on their width by the shrouds, form the passages in the impeller. The vanes have a thickness ranging from 3 to 6 mm and, therefore, reduce the clear area at the inlet circumference. This circumstance constrains the liquid as it enters onto the vanes, and increases the inlet velocity c_1 and changes its direction of flow in comparison with the approach velocity c_0 to the vanes. This change in velocity is evaluated by the inlet crowding factor, $f_1=1.1$ to 1.15.

The velocity of flow drops when the liquid leaves the impeller. This is evaluated by the outlet crowding factor, $f_2=1.05$ to 1.10. The thicker the vanes, as measured along arcs of the inlet and outlet circumferences, and the greater the number of vanes, the less uniformly the velocities will be distributed at the entry into the impeller passages and exit from them, and the higher the losses of head will be in the impeller.

Moreover, a considerable thickness of the vane at inlet and outlet promotes separation of the stream from the vane surfaces which, in turn, impairs the cavitation-proof properties of the pump.

The impeller vanes are designed as thin as is feasible, the inlet edges are rounded and the outlet edges are sharpened. These measures ensure shockless and smooth entry of the liquid onto the vanes under various operating conditions.

An increase in the number of vanes and their length leads to an increase in the loss in head due to the greater area of the surfaces

restricting the flow. The use of too few vanes, however, leads to less uniform distribution of the velocities in the impeller passages, again promoting a loss of head.

The optimal number of vanes can be determined by using the formula developed by Academician G. Proskura:

$$z_v = \frac{Q_{1s}H_1}{2.25\eta_{op}u_2b_r r_2\left[1-\left(\frac{r_1}{r_2}\right)^2\right]} \qquad (212)$$

where Q_{1s} = discharge through the impeller, cu m per sec
H_1 = head developed by the impeller, mH_2O
η_{op} = pump efficiency
b_r = mean radial cross section of the passage, m.

In accordance with the specific velocity of the impeller:
for n_{s1} = 40 to 60, the number of vanes $z_v=9$;
for n_{s1} = 60 to 180, the number of vanes $z_v=8$;
for n_{s1} = 180 to 350, the number of vanes $z_v=6$;
for n_{s1} = 350 to 580, the number of vanes $z_v=5$.

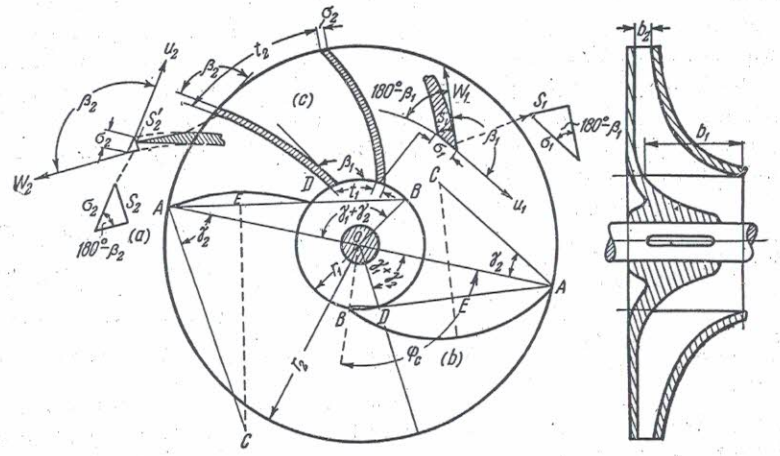


Fig. 115.

To reduce losses in the impeller, the variation in cross section of the passages should be such that $w_1/w_2=1.0$ to 1.15. Modern centrifugal pumps are designed with a ratio within this range.

Many methods may be employed to design the shape of the impeller vanes. The simplest shape is a curve formed of a single circular arc. Given z_v, D_1, D_2, β_1° and β_2° and taking into consideration that the optimal length of the vanes corresponds to a central angle $\varphi_c^\circ \approx$ $\approx$75° to 110° (Fig. 115), we can proceed to construct the vane profile.

The construction of a vane with angles $\gamma_1+\gamma_2>90°$ (Fig. 115*a*) and one with $\gamma_1+\gamma_2<90°$ (Fig. 115*b*) is done in the same manner.

First the inlet and outlet circumferences are drawn. Any point A on the outlet circumference is connected with the centre O and an angle equal to the sum of the angles $180°-\beta_1^\circ=\gamma_1^\circ$ and $180°-\beta_2^\circ=\gamma_2^\circ$ is laid off from the line OA. A line AB is then drawn through the intersection B of the line OB with the inlet circumference, in one case, and through the intersection D of the line OD with the same circumference in the other case. A perpendicular bisector EC is erected to the line AD in the first case (Fig. 115*a*), and to the line AB in the second case (Fig. 115*b*). These perpendicular bisectors are shown with dash lines and at points C they intersect with lines laid off at the angle γ_2° from the lines OA. An arc drawn with C as centre and with the radius AC represents the centre line of the vane. It can be proved that the angles between the tangents to the vane profile and tangents to the inlet and outlet circumferences are equal to β_1° and β_2°, respectively. Involute curves, logarithmic spirals, and other curves can also be used to construct the shape of the vanes.

It must be noted in selecting the vane inlet angle β_1° that large angles lead to lengthening of the vanes and to crowding of the flow at the inlet, while small angles shorten the vanes. Up-to-date designs of centrifugal pumps have vane inlet angles $\beta_1^\circ\approx155°$ to 165°.

The width of the vane, b_1, at inlet and b_2 at outlet can be determined from the formula

$$Q_{1s}=(2\pi r_1-\sigma_1 z_v)\,b_1 c_{1r}=(2\pi r_2-\sigma_2 z_v)\,b_2 c_{2r} \qquad (213)$$

where $\sigma_1=\dfrac{S_1}{\sin(180°-\beta_1^\circ)}$ = vane dimension along an arc of the inlet circumference for a vane thickness S_1

$\sigma_2=\dfrac{S_2}{\sin(180°-\beta_2^\circ)}$ = vane dimension along an arc of the outlet circumference for a vane thickness S_2.

Denoting as t_1 and t_2 the pitch of the vanes on the corresponding circumferences the crowding factors at the inlet and outlet circumferences are, respectively,

$$f_1=\frac{t_1}{t_1-\sigma_1}$$

and

$$f_2=\frac{t_2}{t_2-\sigma_2}$$

Since the peripheral velocities on the line of the inlet circumference, before and after the liquid has entered the impeller, are equal, then

$$c_{0r}=c_{1r}\frac{t_1}{t_1-\sigma_1} \qquad (214)$$

The same is true for the velocities at the outlet circumference of the impeller. Therefore, the radial component of the velocity, after the liquid leaves the impeller, will be

$$c_{3r}=c_{2r}\frac{t_2-\sigma_2}{t_2} \qquad (215)$$

1-26. Recuperating Devices of Centrifugal Pumps

(A) PRINCIPLES OF RECUPERATION (CONVERSION OF DYNAMIC INTO STATIC HEAD)

The velocity of flow when the liquid leaves the impeller may exceed 50 m per sec. A recuperating device is designed to reduce this velocity to a value permissible for flow in pipelines, collecting the liquid and providing favourable conditions for liquid flow after the impeller. Partial conversion of dynamic into static head occurs when the velocity is reduced.

Recuperating devices may be classified as diffuser-, or guide-rings, and volute chambers.

Diffuser-rings, consisting of fixed vanes, are used both for single- and multistage centrifugal pumps. In the latter case, they serve to provide optimal conditions for transfer of the liquid from one stage to the next and also convert part of the kinetic energy into static energy.

Volute chambers are also employed both for single- and multistage pumps. In multistage pumps they are arranged at the outlet of the last stage and are the final recuperating section. A volute chamber may comprise the volute proper and the conical outlet branch (also called the divergent end section).

Let us consider the principle of operation of these two types of recuperating devices. For this purpose we shall assume a chamber free of vanes into which the impeller discharges liquid at an absolute velocity c_3 (Fig. 116*a*). Such chambers are sometimes used in single-impeller low-pressure pumps. Let us assume that upon exit from the chamber the absolute velocity is c_4. The moment of impulse, M_i, for liquid with a mass m passing through the recuperating device in one second will be

$$M_i=m\,(c_4 r_4\cos\alpha_4^\circ-c_3 r_3\cos\alpha_3^\circ)=m\,(c_{4u}r_4-c_{3u}r_3) \qquad (216)$$

If we assume that an inviscid liquid passes through the impeller then the work done by it in passing through the recuperating device is equal to zero, i.e., according to equation (216) we obtain

$$c_{4u}r_4-c_{3u}r_3=0,$$
$$c_{4u}r_4=c_{3u}r_3=c_u r=k_u=\text{const} \qquad (217)$$

Equation (217) expresses the constancy of the moment of momentum for a unit mass of an inviscid liquid, in free vortex flow, in reference to the centre of curvature of the channel and also indicates that the farther an element of liquid is from the centre of rotation, the less its velocity will be.

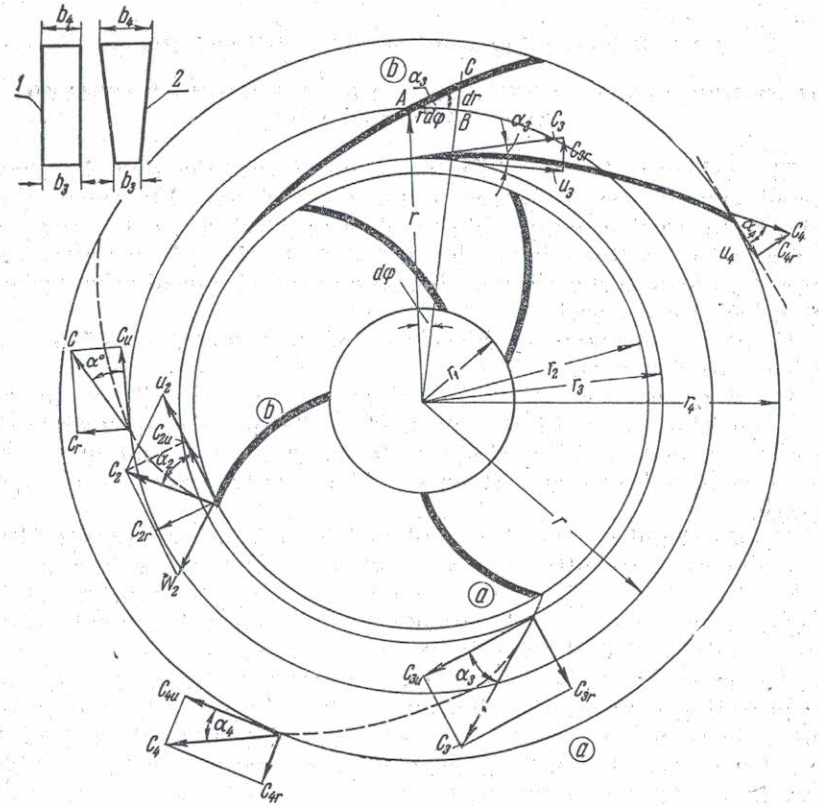


Fig. 116.

From this it follows, on the basis of Bernoulli's equation, that in layers of liquid located farther from the centre of curvature the pressure will be higher than in the nearer layers. As a matter of fact, Bernoulli's equation for flow in a channel in a horizontal plane can be written as

$$\frac{p_3}{\gamma}+\frac{c_{3u}^2}{2g}=\frac{p_4}{\gamma}+\frac{c_{4u}^2}{2g}$$

or

$$\frac{p_4}{\gamma}-\frac{p_3}{\gamma}=\frac{1}{2g}(c_{3u}^2-c_{4u}^2) \qquad (218)$$

According to the equation (217)

$$c_{3u}r_3=k_u \quad \text{or} \quad c_{3u}=\frac{k_u}{r_3}$$

and

$$c_{4u}r_4=k_u \quad \text{or} \quad c_{4u}=\frac{k_u}{r_4}$$

Substituting the values obtained for c_{3u} and c_{4u} into equation (218)

$$\frac{p_4}{\gamma}-\frac{p_3}{\gamma}=\frac{1}{2g}\left(\frac{k_u^2}{r_3^2}-\frac{k_u^2}{r_4^2}\right)=\frac{k_u^2}{2g}\left(\frac{1}{r_3^2}-\frac{1}{r_4^2}\right) \qquad (219)$$

Since $r_4>r_3$, $p_4-p_3>0$.

It thus follows from equation (219) that, to convert dynamic into static head, the flow of liquid leaving the impeller must be discharged into a channel of a shape enabling its velocity to be reduced and its distance from the centre of rotation to be increased.

The shape of the surface of the recuperating device should be such that it promotes shockless flow of the liquid. Consequently, this surface must conform in some degree with the path of the stream of liquid after it leaves the impeller.

Let us determine the path of an element of an inviscid liquid thrown into a channel free of vanes (Fig. 116*b*). Let us assume that at the radius r (point A) the element has a radial velocity c_r and a peripheral velocity c_u, directed at an angle $\alpha°$ to each other. Then

$$\tan\alpha°=\frac{c_r}{c_u}. \qquad (220)$$

According to the equation (217), $c_u=\frac{k_u}{r}$.

Denoting by b the width of the channel (perpendicular to the plane of the drawing), we can express the discharge through the channel per second as

$$Q_s=2\pi rbc_r \quad \text{or} \quad c_r=\frac{Q_s}{2\pi rb}$$

Substituting into equation (220) we obtain

$$\tan\alpha°=\frac{c_r}{c_u}=\frac{\frac{Q_s}{2\pi rb}}{\frac{k_u}{r}}=\frac{Q_s}{2\pi bk_u} \qquad (221)$$

Since the values in equation (221) are constant, angle $\alpha°$, made by the tangent to any circle with the centre O and the tangent to the path of the liquid, is also constant and the path described by the liquid is a logarithmic spiral. Actually, due to the finite number of vanes in the recuperating device, eddy formation and the forces of friction, the path of the elements of liquid differs from a logarithmic spiral. This discrepancy is taken into consideration in design by the introduction of various correction factors.

(B) DIFFUSER-RINGS

The velocity of flow will be reduced in a diffuser-ring if the exit area of its intervane passages is larger than the inlet area. The side walls of the ring (Fig. 116*c*) may be either parallel as in *1* or diverging toward the exit as in *2*.

The finite number of vanes in the impeller influences the redistribution of velocities as the liquid leaves the vanes. This circumstance must be taken into account if we are to ensure shockless entry onto the vanes of the diffuser-ring.

Applying the equation for determining the head of a centrifugal impeller we can write

$$H_t = \frac{H}{\eta_h} = \frac{k_{2cu} u_2 c_2 \cos \alpha_2^\circ}{g} \tag{222}$$

Neglecting the change in head in the annular clearance between the impeller and the diffuser-ring, we can assume that the head at impeller outlet is equal to the head at the entry of the diffuser-ring, then

$$\frac{H}{\eta_h} = \frac{k_{2cu} u_2 c_2 \cos \alpha_2^\circ}{g} = \frac{u_3 c_3 \cos \alpha_3^\circ}{g} \tag{223}$$

Since velocity $u_2 = u_3$ (the peripheral velocity at the inlet to the diffuser-ring) it follows from equation (223) that

$$k_{2cu} c_2 \cos \alpha_2^\circ = c_3 \cos \alpha_3^\circ \tag{224}$$

If leakage through the clearances is neglected, the amounts of liquid passing through the impeller and the diffuser-ring are evidently equal to each other. This means that $c_{2r} = c_{3r}$, i. e.,

$$c_2 \sin \alpha_2^\circ = c_3 \sin \alpha_3^\circ$$

Combining the last two equations we obtain

$$\tan \alpha_3^\circ = \frac{\tan \alpha_2^\circ}{k_{2cu}} \tag{225}$$

The recuperating capacity of the diffuser-ring increases with a decrease in angle α_3°. Since the path of the elements of liquid after they leave the impeller is a logarithmic spiral, the initial section of the diffuser-ring vanes or their full length is made to conform to this curve.

Let us derive the equation for the path of the elements of liquid after they leave the impeller.

We draw two radii with a central angle $d\varphi$ from the centre of the impeller. These radii intercept an arc AB with a radius r on the profile of the diffuser-ring and the line AC on the profile of the vane. If the vane profile conforms with a logarithmic spiral, angle BAC equals α_3° and then

$$\frac{dr}{r d\varphi} = \tan \alpha_3^\circ \quad \text{or} \quad d\varphi = \frac{dr}{r \tan \alpha_3^\circ}$$

Therefore

$$\varphi = \int_{r_3}^{r} \frac{dr}{r \tan \alpha_3^\circ} = \frac{\ln \frac{r}{r_3}}{\tan \alpha_3^\circ} = \cot \alpha_3^\circ \ln \frac{r}{r_3}$$

Converting angle $\varphi°$ into degrees we obtain

$$\varphi° = \frac{180°}{\pi} \cot \alpha_3^\circ \ln \frac{r}{r_3} \tag{226}$$

Using this equation to find values of $\varphi°$ for various values of r, we can plot the points and construct the curve of the diffuser-ring vane.

Next we shall determine angle α_4° at the outlet circumference of the diffuser-ring. Neglecting the difference in thickness of the vane at the inlet and outlet and on the basis of continuity, we can write the following equation for the inlet and outlet circumferences of the diffuser-ring:

$$Q_s = 2\pi r_3 b_3 c_3 \sin \alpha_3^\circ = 2\pi r_4 b_4 c_4 \sin \alpha_4^\circ$$

then

$$\frac{c_4}{c_3} = \frac{b_3 r_3 \sin \alpha_3^\circ}{b_4 r_4 \sin \alpha_4^\circ} \tag{227}$$

Assuming that the shape of the diffuser-ring does not influence the character of the path of elements of liquid passing through it, and on the basis of the theorem of moment of impulse

$$M_u = (c_4 r_4 \cos \alpha_4^\circ - c_3 r_3 \cos \alpha_3^\circ) m = 0$$

Then

$$\frac{c_4}{c_3} = \frac{r_3 \cos \alpha_3^\circ}{r_4 \cos \alpha_4^\circ} \tag{228}$$

Combining equations (227) and (228) we obtain

$$\tan \alpha_4^\circ = \frac{b_3}{b_4} \tan \alpha_3^\circ \qquad (229)$$

The diffuser-ring should have the least possible number of vanes z_{vd}, in comparison with the number of impeller vanes, so as to reduce the loss of head in the ring and to improve its recuperating capacity. Usually, $z_{vd} = z_v \pm 1$, where z_v is the number of impeller vanes. In recent years pumps with other values of z_{vd} have been manufactured.

The radius of the inlet circumference of the diffuser-ring is assumed to be

$$r_3 \approx 1.025 r_2$$

to ensure equalization of velocities after the liquid leaves the impeller and to provide a sufficient clearance between the impeller and the ring.

The outside radius of the diffuser-ring is

$$r_4 = (1.4 \text{ to } 1.8) r_3 \text{ mm}$$

The width of the passage between the fixed vanes is taken to be

$$b_4 = (1.2 \text{ to } 2.0) b_2$$

The ends of the diffuser-ring vanes are usually sharpened or rounded, and the thickness of the initial section depends upon the mechanical properties of the metal. It may range from 1 to 10 mm.

The divergence angle of the intervane passages is made from 8° to 12° to avoid separation of the flow from the walls. As a rule, the outlet velocity from the diffuser-ring does not exceed 15 m per sec.

The recuperating capacity of a diffuser-ring can be determined from Bernoulli's equation relating the inlet and outlet cross-sectional areas.

Neglecting the nonuniform distribution of velocities in the passages, we can write

$$\frac{p_3}{\gamma} + \frac{c_3^2}{2g} = \frac{p_4}{\gamma} + \frac{c_4^2}{2g} + \zeta_{dr} \frac{(c_3 - c_4)^2}{2g}$$

Then

$$\frac{p_4}{\gamma} - \frac{p_3}{\gamma} = \frac{c_3^2 - c_4^2}{2g} - \zeta_{dr} \frac{(c_3 - c_4)^2}{2g} = h_{dr} \qquad (230)$$

where h_{dr} = increase in static head in the diffuser-ring

$\zeta_{dr} \approx 0.15$ to 0.20 = coefficient of hydraulic resistance of the diffuser-ring, referred to head losses due to shock according to Borda's theorem.

The walls of the intervane passages should be finished as smooth as possible to reduce the loss of head due to friction.

(C) VOLUTE CHAMBER

Until recently, a diffuser-ring was considered as good a recuperating device as a volute chamber or even better.

This was due to the fact that the channels of a volute are not accessible for machining and their shape, surface finish and dimensions depend upon the foundry techniques employed in their manufacture.

From the point of view of the hydraulic principles involved, however, a volute channel can be made with a much more favourable shape than the rectangular shape of the passages in a diffuser-ring. This can reduce shock and eddy formation losses to a minimum, notwithstanding the higher friction losses in the volute.

Volute chambers were commonly used for low-pressure single-stage pumps because of the reasons mentioned above and because they can be conveniently arranged in the design of such pumps.

A diffuser-ring, being rectangular in shape, is less expedient for liquid flow from the point of view of hydraulics, but it takes up less space and can be machined. Consequently, its friction losses are less than for a volute. On these grounds, diffuser-rings were usually applied in high-pressure and multistage pumps.

With the development of up-to-date foundry techniques, enabling smooth surfaces of accurate shape to be obtained in the channels, volutes are being more and more frequently resorted to in the design of multistage pumps. Pump manufacturing practice shows that in many cases pumps with diffuser-rings have a lower efficiency than pumps with a properly designed volute chamber.

In its principle of operation a volute does not differ from a diffuser-ring. An element of liquid entering the volute *1* (Fig. 117) passes to the exit (*a-VIII*). Since the cross section of the volute is designed so that the element gradually moves away from the centre in the radial direction, thereby losing velocity, recuperation (conversion of dynamic into static head) takes place along its whole path from entry to exit.

As a rule, volutes are designed for a reduction of velocity in the range from 20 to 25 per cent. The amount of flow accommodated by the volute increases gradually from the smallest cross section *2* (tongue) to the exit. For this reason, and to promote a gradual retardation of the liquid, the volute chamber has a diverging cross section.

The recuperating action of the volute can be intensified by equipping it with a conical outlet branch *3* which diverges at an angle from 8° to 10°. The velocity of the liquid discharged from this branch is reduced to 5 or 6 m per sec. Recuperation takes place chiefly while the liquid passes through the conical outlet branch. Further reduction of velocity of flow to that in the discharge line (2 to 4 m per sec)

is accomplished in the divergent end section 4 which connects the pump to the line.

At low outlet velocities from the conical branch, the latter may have quite a large cross section. Most extensively used are volutes of round, rectangular and trapezoidal cross section.

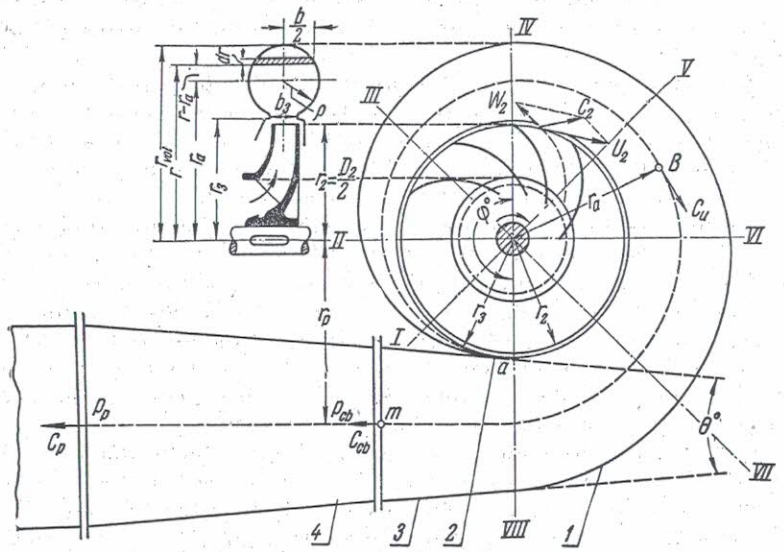


Fig. 117

If the losses of head in the volute are neglected, it can be affirmed that the flow of liquid through it conforms to equation (217). Let us consider the method used to design volutes of round cross section.

First we shall divide the volute chamber by planes into a number of cross sections (Fig. 117) and consider the cross section located at an angle of $\varphi°$ to the tongue. Accepting the notation shown on the view of this section we can say that the liquid discharge, dQ_φ, through an element of area of the section with a width b and a height dr for a flow velocity of c_u is

$$dQ_\varphi = c_u b dr = \frac{k_u b}{r} dr$$

The liquid discharge through the whole cross section being considered is

$$Q_\varphi = \int_{r=a-\varrho}^{r=a+\varrho} \frac{k_u b\, dr}{r} \qquad (231)$$

On the other hand, if Q_s is the discharge per second of the impeller, then its discharge over the section bounded by angle $\varphi°$ will be

$$Q_\varphi = \frac{Q_s}{360}\varphi° \qquad (232)$$

Combining equations (231) and (232) and solving for $\varphi°$ we obtain

$$\varphi° = \frac{360}{Q_s}\int_{r=a-\varrho}^{r=a+\varrho} \frac{k_u b\, dr}{r}$$

According to Fig. 117

$$\left(\frac{b}{2}\right)^2 = \rho^2 - (r-r_a)^2 \quad \text{or} \quad b = 2\sqrt{\rho^2-(r-r_a)^2}$$

then

$$\varphi° = \frac{360k_u}{Q_s}\int_{r=a-\varrho}^{r=a+\varrho} 2\sqrt{\rho^2-(r-r_a)^2}\frac{dr}{r} = \frac{720k_u}{Q_s}\int_{r=a-\varrho}^{r=a+\varrho}\sqrt{\rho^2-(r-r_a)^2}\frac{dr}{r}$$

After integrating the last expression we can write

$$\varphi° = \frac{720}{Q_s}k_u\pi\left(r_a-\sqrt{r_a^2-\rho^2}\right) = \frac{720}{Q_s}k_u\pi\left[(r_3+\rho)-\sqrt{(r_3+\rho)^2-\rho^2}\right] \qquad (233)$$

Assuming various values of radius ρ and laying off double their value radially from a circle with a radius r_3 at the corresponding calculated angles $\varphi°$, we can obtain the directrix of the volute, from which the volute itself can be drawn. It proves more convenient, however, to use an equation in which values of the bounding angle $\varphi°$ are assumed and the corresponding values of ρ are calculated. This can be done by transforming equation (233), assuming that

$$\frac{720}{Q_s}k_u\pi = \chi$$

then

$$\varphi° = \chi\left(r_a-\sqrt{r_a^2-\rho^2}\right)$$

After dividing both sides of the equation by χ, transferring r_a to the left side and squaring both sides. we obtain

$$\left(r_a-\frac{\varphi°}{\chi}\right)^2 = r_a^2-\rho^2$$

$$r_a^2-2\frac{\varphi°}{\chi}r_a+\left(\frac{\varphi°}{\chi}\right)^2 = r_a^2-\rho^2$$

$$\rho^2-2\frac{\varphi°}{\chi}r_a+\left(\frac{\varphi°}{\chi}\right)^2 = 0$$

substituting for r_a

$$\rho^2-2\frac{\varphi^\circ}{\chi}(r_3+\rho)+\frac{\varphi^{\circ 2}}{\chi^2}=0$$

$$\rho^2-2\frac{\varphi^\circ}{\chi}\rho+\left(\frac{\varphi^{\circ 2}}{\chi^2}-2\frac{\varphi^\circ}{\chi}r_3\right)=0$$

$$\rho=\frac{2\frac{\varphi^\circ}{\chi}}{2}+\sqrt{\left(\frac{2\frac{\varphi^\circ}{\chi}}{2}\right)^2-\left(\frac{\varphi^{\circ 2}}{\chi^2}-2\frac{\varphi^\circ}{\chi}r_3\right)}=\frac{\varphi^\circ}{\chi}+\sqrt{2\frac{\varphi^\circ}{\chi}r_3} \quad (234)$$

Assuming various values of angle φ° and calculating the radii ρ for each bounding angle we can construct the directrix of the volute.

Calculations can be most conveniently carried out in tabular form (Table 31).

Table 31

Cross section No.	φ°	$\frac{\varphi^\circ}{\chi}$	$2\frac{\varphi^\circ}{\chi}$	$2\frac{\varphi^\circ}{\chi}r_3$	$\sqrt{2\frac{\varphi^\circ}{\chi}r_3}$	ρ	$r_3+\rho$	$r_3+2\rho=r_{vol}$	Notes

The outside r_{vol} and inside r_3 radii of the volute are

$$r_{vol}=(1.3 \text{ to } 1.5)\,r_2 \quad (235)$$

$$r_3=(1.02 \text{ to } 1.05)\,r_2 \quad (236)$$

The width of the opening in the volute from the impeller is taken to be

$$b_3\approx b_2+0.025r_2 \quad (237)$$

to make provision for axial shift of the impeller.

The increase in static head, Δh_{st}, can be found from Bernoulli's equation for the outlet sections of the impeller and conical branch.

The equation for a stream of liquid passing along the centre line *abm* of the volute is

$$\frac{p_2}{\gamma}+\frac{c_2^2}{2g}=\frac{p_{cb}}{\gamma}+\frac{c_{cb}^2}{2g}+\zeta_{vol}\frac{c_{cb}^2}{2g}$$

Then

$$\Delta h_{st}=\frac{p_{cb}-p_2}{\gamma}=\frac{c_2^2}{2g}-(1+\zeta_{vol})\frac{c_{cb}^2}{2g}, \quad (238)$$

where ζ_{vol} = coefficient of hydraulic resistance of the volute (see Table 32).

Table 32

Number of impeller vanes, z_v	6	8	10	12
Value of ζ_{vol} for $\beta_2^\circ=150^\circ$	0.12	0.17	0.22	0.255

The increment of static head recuperated in the divergent end section can be determined from Bernoulli's equation for the inlet and outlet of the end section. Thus

$$\Delta h_{dst}=\frac{p_{cb}-p_p}{\gamma}=\frac{c_{cb}^2}{2g}-(1+\zeta_{des})\frac{c_p^2}{2g} \quad (239)$$

where $\zeta_{des}\approx 0.15$ = coefficient of hydraulic resistance of the divergent end section.

The taper angle θ° of the divergent end section should not exceed 15° as otherwise the stream of liquid may separate from the walls.

1-27. Axial Hydraulic Thrust and Impeller Balancing

During centrifugal pump operation, the liquid entering the inlet connection 7 (Fig. 118) creates a pressure p_1 in the inlet chamber *8* which is lower than the pressure p_2 at impeller outlet *10*.

If there was no rotary motion of the liquid in the clearance spaces *1* and *2*, due to friction between the impeller shrouds and the liquid, the pressure in these spaces would be equal to p_2.

As a result of the viscosity of the liquid, its rotary motion is established in spaces *1* and *2* about the axis of impeller rotation at an angular velocity equal to about one half of that of the shrouds. This motion and the consequent centrifugal force leads to a reduction of pressure in spaces *1* and *2* in the direction towards the impeller axis.

An approximate pressure distribution diagram illustrating this point is shown in Fig. 118.

The pressure of the liquid acting on annular areas with a width D_2-D_0 on the right- and left-hand shrouds is counterbalanced since the forces are equal in magnitude and opposite in direction.

Thus, the remaining unbalanced forces R_1 and R_2 act from the right and left on an annular area with a width D_0-d_{sh}.

Denoting the pressure in the inlet connection 7 as p_1 and that in space *2* as p_2, we obtain as a first approximation that

$$R'=R_2-R_1=\frac{\pi}{4}(p_2-p_1)(D_0^2-d_{sh}^2) \text{ kg}$$

As the liquid enters the impeller passages its flow is diverted from the axial to the radial direction. This results in an axial thrust R_3 from left to right due to the pressure of the liquid on the curved

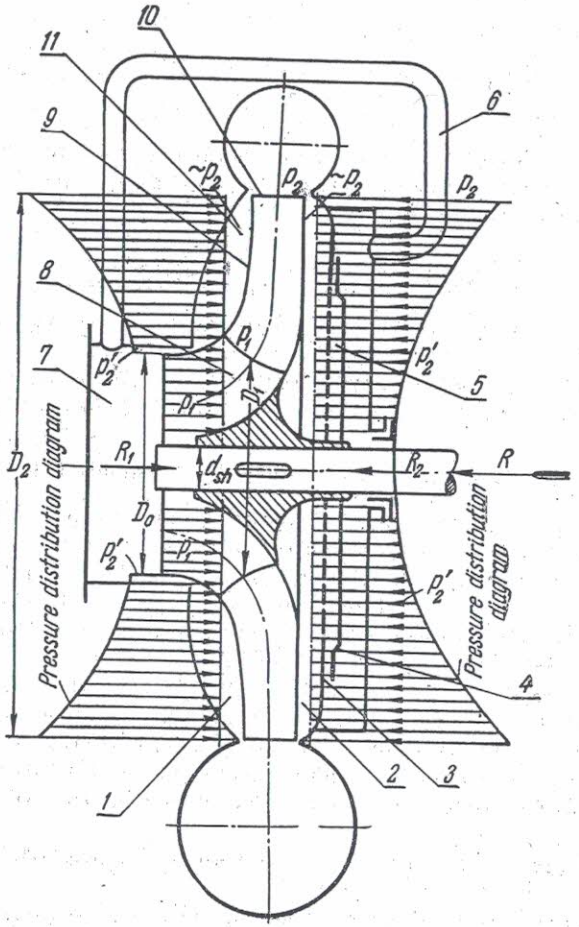


Fig. 118.

shroud of the impeller. Applying the law of momentum for liquid flowing with a velocity c_0 before it enters the impeller, we obtain

$$R_3 = mc_0 = \frac{\gamma Q_s}{g} c_0$$

Therefore, the resultant axial thrust tending to shift the impeller to the right will be

$$R = R' - R_3 = \frac{\pi}{4}(p_2 - p_1)(D_0^2 - d_{sh}^2) - \frac{\gamma Q_s}{g} c_0$$

In multistage pumps the total axial thrust is equal to the sum of the axial thrusts applied to the separate impellers, and may reach several tons.

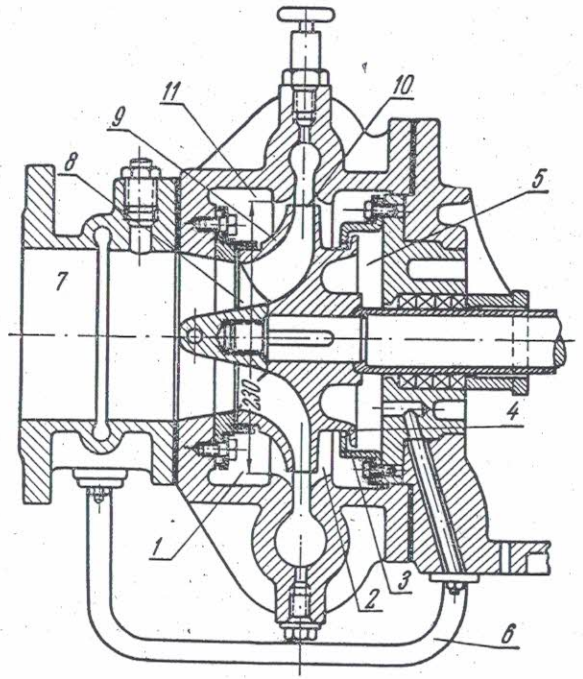


Fig. 119.

Pump impellers are either relieved of axial thrust or the forces are balanced.

The following means are used to balance the axial thrust:

(1) balancing devices of the hydraulic type;
(2) thrust bearings;
(3) parallel double admission of the liquid to the impeller.

The axial thrust on impellers is relieved by the provision of balancing holes or pipes to equalize the pressures on the two sides.

Figs. 118 and 119 illustrate the principle of one hydraulic balancing device and the construction of a pump incorporating this device. From the inlet connection 7 the liquid being pumped passes through

the inlet chamber *8* of impeller *9*, through the impeller passages and is discharged through the outlet circumference *10*. Through clearances between the impeller and the casing, the liquid enters the clearance spaces *1* and *2*. Balancing disk *4* is secured on the hub of impeller *9*. During normal steady operation of the pump a gap of definite size is maintained between disk *4* and its stationary seat *3*. Through this gap liquid passes into balance chamber *5* (beyond disk *4*) and from there through pipe *6* to the inlet connection *7*.

The area of disk *4* and the size of the gap between it and seat *3* are designed so that the difference in pressure in clearance space *2* and chamber *5* counterbalances the axial thrust R_1 which tends to shift the impeller towards the right, thereby preventing such a shift.

Let us assume that equilibrium has been impaired and the impeller shifts to the left. In this case, the gap between the disk and its seat is reduced and the pressure drops in chamber *5*. Then the pressure in space *2* will return the impeller to its initial balanced position.

If the impeller shifts to the right, the pressure in chamber *5* is increased, again returning the impeller to its initial position. Normal axial shift of the balancing disk ranges from 0.5 to 1 mm.

Thrust bearings are used to balance the axial thrust in the pump illustrated in Fig. 110. A pump with double admission of the liquid is shown in Fig. 108.

The use of balancing holes *19* in conjunction with sealing rings for relieving axial thrust is shown in Fig. 109. Since the holes *19* connect the space at the right side of the shroud with the inlet chamber, they both have the same pressure and no axial thrust appears.

It must be noted, however, that it is not possible to completely balance the axial thrust by employing double admission impellers, or to completely relieve it by using balance holes. This is due to the impossibility of obtaining absolutely identical shapes of the surfaces in the impeller passages and identical grades of surface finish. Therefore, even pumps with double admission are equipped with light thrust bearings.

1-28. Pump Losses, Efficiency and Power Input

Losses in pumps may be of three types: hydraulic, volumetric and mechanical.

Hydraulic losses reduce the head developed by the impeller. They are made up of losses in overcoming friction forces due to the viscosity of the liquid, losses from eddy formation and impact, or shock, losses.

Friction losses depend upon the surface roughness in the passages and constitute only an insignificant part of the total losses if the surfaces are properly machined.

Eddy losses are due to breaking away of the stream from the passage surfaces, flow of liquid from a high-pressure to a low-pressure region in the passages, and flow over projecting parts with sharp angles during inlet and exit from the impeller vanes.

Impact occurs when the liquid meets an obstruction or when its flow velocity is changed either in magnitude or direction.

Volumetric losses consist of the leakages of liquid through clearances from high-pressure to low-pressure regions, and reduce the pump capacity.

The most significant volumetric losses are:

(1) Leakage through the front sealing ring of the impeller due to internal circulation about the impeller shroud from clearance space *11* (Figs. 118 and 119) into the inlet connection *7*.

(2) Leakage through the balancing disk or balancing holes.

(3) Leakage through the shaft packing or through the hub seals in multistage pumps.

Mechanical losses reduce the power transmitted to the pump impeller so that the energy imparted to the liquid is less than that developed on the pump shaft.

Mechanical losses consist of:

(1) Rotation losses due to friction of the external surfaces of the impeller shrouds and balancing disks against the liquid surrounding them.

(2) Friction losses in the shaft bearings.

(3) Friction losses in the shaft stuffing boxes.

The hydraulic losses are evaluated by the hydraulic efficiency:

$$\eta_h = \frac{H}{H_t} = \frac{H}{k_{2cu} H t\infty} = \frac{\gamma Q_s H}{\gamma Q_s H_t} \cong 0.75 \text{ to } 0.95 \qquad (240)$$

Volumetric losses are taken into account by the volumetric efficiency η_v which is the ratio of the actual discharge per second, Q_s, to the theoretical discharge, Q_{ts}, thus

$$\eta_v = \frac{Q_s}{Q_{ts}} \approx 0.94 \text{ to } 0.995 \qquad (241)$$

The variation of volumetric efficiency with the specific velocity is shown in Table 33.

Table 33

n_{s1}	60 to 100	100 to 150	150 to 220
η_v	0.94 to 0.97	0.97 to 0.99	0.98 to 0 995

The ratio of the useful power $N_u=\gamma Q_s H$ kg-m per sec to the indicated power $N_i=\gamma Q_{ts} H_t$ kg-m per sec applied to the pump impeller is called the internal efficiency η_i of the pump:

$$\eta_i=\frac{\gamma Q_s H}{\gamma Q_{ts} H_t}=\eta_v \eta_h=0.70 \text{ to } 0.9 \qquad (242)$$

The losses due to mechanical friction of the moving parts of the pump and to friction of the shroud surfaces on the liquid are taken into account by the mechanical efficiency.

If N kg-m per sec is the power developed on the pump shaft, then

$$\eta_m=\frac{N_i}{N}=\frac{\gamma Q_{ts} H_t}{N}=0.9 \text{ to } 0.97 \qquad (243)$$

The overall efficiency of the pump takes into consideration all the losses, and is equal to

$$\eta_{op}=\frac{\gamma Q_s H}{N}=0.63 \text{ to } 0.84 \qquad (244)$$

After solving equation (243) for N, substituting this value in (244) and combining with equation (242) we obtain

$$\eta_{op}=\frac{\gamma Q_s H}{N}=\frac{\gamma Q_s H}{\frac{N_i}{\eta_m}}=\frac{\gamma Q_s H}{\gamma Q_{ts} H_t}\eta_m=\eta_i \eta_m=\eta_v \eta_h \eta_m \qquad (245)$$

Carefully manufactured high-capacity pumps may have an overall efficiency η_{op} up to 0.9.

Approximate values of overall pump efficiency are listed in Table 34 as a function of the value

$$A=\sqrt[3]{\frac{Q_{sl} n}{1{,}000}}$$

where Q_{sl}=pump discharge per second, in litres.

Table 34

Multistage pumps								
$A=\sqrt[3]{\frac{Q_{sl} n}{1{,}000}}$	5	7.5	10	12.5	15	25	35	50
η_{op}	0.60	0.68	0.725	0.75	0.77	0.80	0.82	0.84

Single-stage pumps

$A=\sqrt[3]{\frac{Q_{sl} n}{1{,}000}}$	5	10	15	20	30	40	80	—
η_{op}	0.65	0.75	0.785	0.82	0.86	0.88	0.9	—

1-29. Determining the Principal Dimensions of the Pump Impeller

The principal dimensions of the pump impeller (Fig. 120) must be selected so as to provide the required head, H, and capacity, Q, at a rotational speed n conforming to that of a standard motive unit. It follows from what has been set forth previously that these values

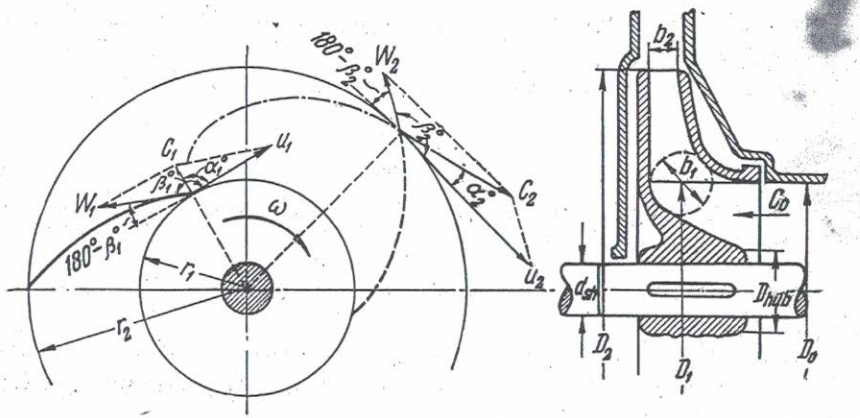


Fig. 120.

depend upon the liquid velocities at the inlet and outlet circumferences of the impeller, and the angles of the velocity diagrams at these circumferences.

The values of the most expedient velocities can be expressed in terms of the actual head developed by the impeller, and its specific velocity by means of the formulas

$$u=k_u\sqrt{2gH}$$
$$c=k_c\sqrt{2gH}$$

where $k_u=F_1(n_{s1})$=peripheral velocity factor
$k_c=F_2(n_{s1})$=absolute velocity factor.

The specific velocity of the impeller is

$$n_{s1}=\frac{3.65n\sqrt{Q_s}}{\sqrt[4]{H^3}}$$

The type of pump is selected according to the specific velocity (see Fig. 104).

The design capacity of the pump is defined as the actual amount of liquid passing through the impeller and is equal to the theoretical discharge per second

$$Q_{ts}=\frac{Q}{3{,}600\eta_v} \text{ cu m per sec}$$

where η_v=volumetric efficiency taken from Table 33.

The overall efficiency of the pump, η_{op}, is found in Table 34 depending upon the value $A=\sqrt[3]{\frac{Q_{sl}n}{1{,}000}}$. Here Q_{sl} is the impeller discharge, litres per sec.

The power required on the pump shaft is

$$N=\frac{\gamma Q_s H}{75\eta_{op}} \text{ hp}$$

The diameter of the pump shaft is calculated on the basis of its torsional strength. Thus

$$d_{sh}=A_{tw}\sqrt[3]{\frac{N}{n}} \text{ m}$$

where A_{tw}=a factor depending upon the permissible angle of twist, φ°_{tw}, per 1 m of length (see Table 35), ranging from 0.25° to 0.5° for pumps.

Table 35

Permissible angle of twist, φ°_{tw}	0.25	0.5	1	1.5	2	2.5
Factor A_{tw}	12	10	8.5	7.7	7.2	6.8

The diameter of the impeller hub depends upon the distance the key projects from the shaft and upon the design of the hub:

$$D_{hub}=(1.205 \text{ to } 1.4)\,d_{sh}$$

where d_{sh}=shaft diameter, m.

The diameter of the inlet connection is determined from the principle of continuity of flow:

$$D_c=\sqrt{\frac{4Q_{ts}}{\pi c_c}} \text{ m}$$

where c_c= velocity of flow in the inlet connection, m per sec

$c_c\approx$ 2 to 4 m per sec for pumps installed above the level of the liquid being pumped

$c_c\approx$5 to 6 m per sec for pumps operating with negative lift.

The velocity of the liquid in the inlet chamber of the impeller is found from the formula

$$c_0=k_{c_0}\sqrt{2gH} \text{ m per sec}$$

where k_{c0}= (0.058 to 0.06) $n_{s1}^{2/3}$=impeller entry velocity factor.

We determine the diameter of the inlet circumference of the impeller by writing the continuity equation for the area of the inlet opening, or eye:

$$Q_{ts}=c_0\frac{\pi}{4}(D_0^2-D_{hub}^2) \text{ cu m per sec}$$

then

$$D_0=\sqrt{\frac{4Q_{ts}}{\pi c_0}+D_{hub}^2} \text{ m}$$

The diameter of the inlet circumference of the impeller vanes is usually equal to the preceding value. The radial velocity of liquid entry onto the vanes is

$$c_{1r}=k_{c_{1r}}\sqrt{2gH} \text{ m per sec}$$

where $k_{c_{1r}}$=radial velocity factor (see Table 36).

Table 36

n_{s1}	60	100	150	200	220
$k_{c_{1r}}$	0.125	0.125	0.128	0.136	0.142

After determining the number of impeller vanes from n_{s1} and assigning their pitch, t_1, and thickness, S_1, we can find the impeller vane inlet crowding factor

$$f_1=\frac{t_1}{t_1-\sigma_1}$$

Upon radial entry into the impeller, the absolute entry velocity c_1 onto the vanes is equal to the radial component, c_{1r}. Then, taking

into consideration the crowding of the inlet section by the thickness of the vanes, we can write

$$c_1 = \frac{c_0}{f_1} = c_{1r}$$

Applying the continuity principle and determining the width, b_1, of the vanes at entry when $D_0 = D_1$, we obtain

$$Q_{ts} = \pi D_0 b_1 c_0 = \pi D_1 b_1 c_1 = \pi D_0 b_1 c_{1r} \text{ cu m per sec}$$

and

$$b_1 = \frac{Q_{ts}}{\pi D_0 c_0} = \frac{Q_{ts}}{\pi D_1 c_1} = \frac{Q_{ts}}{\pi D_1 c_{1r}} \text{ m}$$

The peripheral entry velocity onto the vanes may be found from the following formulas:

$$u_1 = \frac{\pi D_1 n}{60} \text{ m per sec and } u_1 = k_{u1}\sqrt{2gH} \text{ m per sec}$$

where $k_{u1} = 0.0244 n_{s1}^{2/3}$ = peripheral velocity factor at the inlet circumference of the vanes.

The direction of the relative entry velocity onto the vanes is found from the equation

$$\tan (180° - \beta_1^\circ) = \frac{c_1}{u_1}$$

The magnitude of the relative entry velocity is

$$w_1 = \frac{c_{1r}}{\sin \beta_1^\circ}$$

The liquid velocity at the outlet circumference of the impeller is

$$u_2 = k_{u2}\sqrt{2gH} \text{ m per sec}$$

where $k_{u2} = 1 + 0.1\left(\frac{n_{s1}}{100} - 1\right)$ = peripheral exit velocity factor.

The diameter of the outlet circumference is

$$D_2 = \frac{60\, u_1}{\pi n} \text{ m}$$

An increase in the velocity, c_{2r}, with which the liquid leaves the impeller leads to an increase in pump losses, since this velocity is perpendicular to the velocity of flow in the collecting chamber (volute) and is almost completely lost in shock upon the merging of the two streams of liquid. A reduction in c_{2r} affects the optimum design ratios of the impeller. Pump tests and theoretical investigations have shown that c_{2r} should equal c_{1r} which proves to be the most expedient exit velocity from the impeller.

It follows from the equation for the head developed by the pump that if the relative entry velocity, w_1, is considerably higher than w_2 at the outlet, there will be an increase in static head due to the conversion of velocity head into static head, and pump efficiency will be raised. Moreover, to obtain a radial entry (Fig. 120) a sharper vane entry angle must be assigned and, consequently, a higher relative velocity, w_1.

On the other hand, an increase in the velocity with which the liquid approaches the vane edge leads to higher friction losses. Retardation of the flow in the impeller passages, as is shown by investigations, promotes the formation of eddies and cavitation zones because the flow does not fill the whole cross section. Moreover, the conditions of cavitationless operation of the pump stipulate low inlet velocities to the impeller. Thus, to reduce the losses when the liquid passes through the impeller the relative velocity should be either accelerated or constant. It proves difficult to design an impeller with accelerated relative velocity and, in any case, this would lead to a reduction in the head developed by the pump. The best solution is to assign a constant relative velocity of the liquid passing through the impeller, i.e., $w_1 = w_2 = w$.

The outlet angle, β_2°, of the relative velocity of the liquid leaving the impeller vanes is found from the equation

$$\sin (180° - \beta_2^\circ) = \frac{c_{2r}}{w_2}$$

Table 37 shows the approximate variation of angle β_2° with n_{s1}.

Table 37

n_{s1}	40	100	150	200	250	300
β_2°	144	150	156	160	165	165

The width of the outlet edge of the vane is

$$b_2 = \frac{Q_{ts}}{\pi D_2 c_{2r} f_2} = \frac{Q_{ts}}{\pi D_2 c_{2r}} \frac{t_2 - \sigma_2}{t_2}$$

where f_2 = impeller vane outlet crowding factor.

1-30. Centrifugal Pump Operation

(A) OPERATIONAL CHARACTERISTICS OF PIPELINES AND CENTRIFUGAL PUMPS

The relation between the volume of liquid passing through a given pipeline in unit time and the resulting loss in head is called the *pipeline characteristic.*

The losses of head, H_{pl}, in a pipeline can be conventially divided into static, H_{pls}, and dynamic, H_{pld}, losses.

By static losses of head are meant the head required to lift the liquid being pumped from the free surface to the outlet hole of the discharge line, and the head required to overcome the pressure in the tank into which the liquid is pumped.

The head required to overcome all the hydraulic resistances in the pipeline will be called the dynamic loss of head. Thus, the total head, H_{pl}, lost in the pipeline will be

$$H_{pl}=H_{pls}+H_{pld} \tag{246}$$

If w_d is the summary referred coefficient of resistance of the pipeline attributed to the velocity c_{pl} of flow in any part of the pipeline with a clear section f_{pl}, then

$$H_{pld}=w_d\frac{c_{pl}^2}{2g}=\frac{w_d}{2g}\left(\frac{Q_s}{f_{pl}}\right)^2 \tag{247}$$

where Q_s=discharge through the pipeline, cu m per sec.

Combining equations (246) and (247) we obtain

$$H_{pl}=H_{pls}+\frac{w_d}{2g}\left(\frac{Q_s}{f_{pl}}\right)^2=H_{pls}+k_{pl}Q_s^2 \tag{248}$$

where $k_{pl}=\frac{w_d}{2gf_{pl}^2}$=pipeline constant.

This equation is represented graphically in Fig. 121 by the lines *I*, *II*, *III*, *IV* and *V*. If there is no static head ($H_{pls}=0$) then the dot-and-dash lines *1*, *2*, *3*, *4* and *5* represent the equation.

The larger the value of the referred coefficient of resistance of the pipeline, the steeper the curves representing its characteristics will be.

Pump characteristics, in conjunction with pipeline system characteristics, serve to determine optimum conditions of pump operation, to find the most expedient field of application and to select pumps according to the head, capacity, efficiency and power consumption. A pump characteristic is a functional relation of two parameters determining pump operation when a third parameter is maintained constant. Pump characteristics, in the same manner as parameters, are classified as theoretical, design and actual.

These characteristics may be:

(1) Main characteristic curves plotted for a variable rotational speed of the impeller.

(2) Working characteristic curves plotted for a constant impeller speed.

(3) Universal characteristic curves which are a combination of the main and working characteristics.

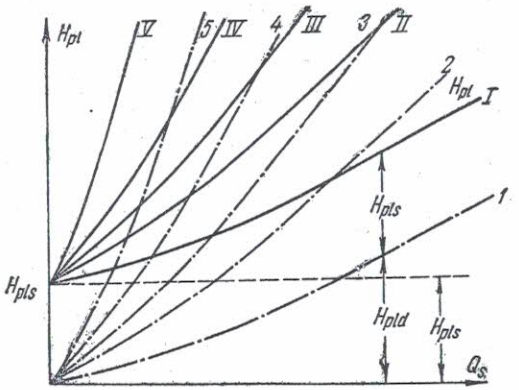


Fig. 121.

The principle pump characteristic is the main characteristic curve showing the relation between the head, H, and the capacity, Q, for variable pump shaft speeds. It is often called the head-discharge characteristic.

First we shall consider the theoretical characteristics of a pump (Fig. 122).

If the impeller rotates at n rpm, we can write, neglecting the thickness of the vanes, that the theoretical discharge per second and the theoretical head for an infinite number of vanes will be

$$\left.\begin{aligned} Q_{ts\infty}&=2\pi r_1 b_1 c_{1r}=2\pi r_2 b_2 c_{2r}\\ H_{t\infty}&=\frac{u_2 c_{2u}}{g}\ \text{mH}_2\text{O}\\ N_{t\infty}&=\frac{Q_{ts\infty}\gamma H_{t\infty}}{75}\ \text{hp}\end{aligned}\right\} \tag{249}$$

When the speed of the pump is changed from n to n', the angles between the velocities of the liquid at entry onto and exit from the impeller are not changed and, therefore, the new velocity diagram

will be similar to the previous one, thus

$$\left.\begin{aligned} Q'_{ts\infty} &= 2\pi r_1 b_1 c'_{1r} = 2\pi r_2 b_2 c'_{2r} \\ H'_{t\infty} &= \frac{u'_2 c'_{2u}}{g} \ \text{mH}_2\text{O} \\ N'_{t\infty} &= \frac{Q'_{ts\infty}\gamma H'_{t\infty}}{75} \ \text{hp} \end{aligned}\right\} \qquad (250)$$

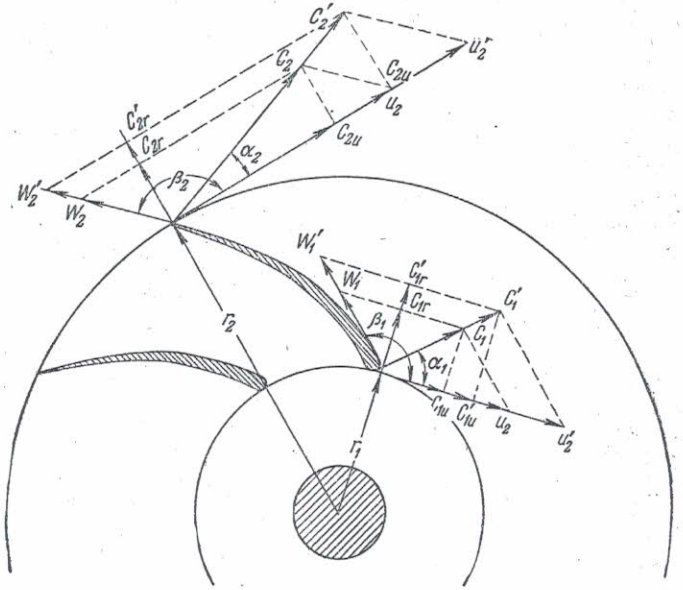


Fig. 122.

Since the magnitude of the velocities varies proportionally to the rotational speed

$$\frac{c'_{1r}}{c_{1r}} = \frac{c'_{2r}}{c_{2r}} = \frac{u'_2}{u_2} = \frac{c'_{2u}}{c_{2u}} = \frac{n'}{n} = \Delta n \qquad (251)$$

On the basis of equations (249), (250) and (251) we obtain

$$\frac{Q_{ts\infty}}{Q_{ts\infty}} = \frac{n'}{n} = \Delta n \text{ or } Q_{ts\infty} = Q_{ts\infty}\frac{n'}{n} = Q_{ts\infty}\Delta n \qquad (252)$$

$$\frac{H'_{t\infty}}{H_{t\infty}} = \frac{(n')^2}{n^2} = \Delta n^2 \text{ or } H'_{t\infty} = H_{t\infty}\frac{(n')^2}{n^2} = H_{t\infty}\Delta n^2 \qquad (253)$$

$$\frac{N'_{t\infty}}{N_{t\infty}} = \frac{(n')^3}{n^3} = \Delta n^3 \text{ or } N'_{t\infty} = N_{t\infty}\frac{(n')^3}{n^3} = N_{t\infty}\Delta n^3 \qquad (254)$$

Combining equations (252), (253) and (254) we obtain

$$\frac{(Q'_{ts\infty})^2}{Q^2_{ts\infty}} = \frac{H'_{t\infty}}{H_{t\infty}} \text{ and } \left(\frac{Q'_{ts\infty}}{Q_{ts\infty}}\right)^3 = \frac{N'_{t\infty}}{N_{t\infty}} \qquad (255)$$

The ratio of the impeller shaft torques, when the rotational speed is changed, can be found from the equation:

$$\frac{N'_{t\infty}}{N_{t\infty}} = \frac{(n')^3}{n^3} = \frac{M'n'}{Mn} \text{ or } \frac{M'}{M} = \frac{(n')^3}{n^3}\frac{n}{n'} = \frac{(n')^2}{n^2} = \Delta n^2 \qquad (256)$$

It should be noted that the specific velocity of the given impeller is not changed with the rotational speed. In fact

$$\frac{n'_{s1}}{n_{s1}} = \frac{3.65\dfrac{n'\sqrt{Q'_s}}{\sqrt[4]{(H')^3}}}{3.65\dfrac{n\sqrt{Q_s}}{\sqrt[4]{H^3}}} = \frac{n'\sqrt{Q_s\left(\dfrac{n'}{n}\right)}\sqrt[4]{H^3}}{n\sqrt{Q_s}\sqrt[4]{H^3\left[\left(\dfrac{n'}{n}\right)^2\right]^3}} = 1 \qquad (257)$$

It has been established by research that the derived ratios change only slightly under actual impeller operation, if the rotational speed has been changed by not more than 1.5 or 2 times. Consequently, these ratios can be employed for computing actual capacities, heads and powers.

When the rotational speed is changed by more than 1.5 to 2 times it is necessary to take into consideration the change in the overall pump efficiency η_{op} in the calculation of actual characteristics. This change is due to the changes in the mechanical and hydraulic losses which vary in the following manner:

$$\frac{1-\eta_{op}}{1-\eta'_{op}} \approx \left(\frac{n'}{n}\right)^{0.2} \approx \frac{1-\eta_h}{1-\eta'_h}$$

The actual main characteristics of the pump can be expressed as

$$\frac{Q'}{Q} = \frac{n'}{n}\ \frac{\eta'_h H'}{\eta_h H} = \left(\frac{n'}{n}\right)^2 \text{ and } \frac{N'\eta'_{op}}{N\eta_{op}} = \left(\frac{n'}{n}\right)^3$$

The hydraulic efficiency can be found from a formula proposed by A. Lomakin:

$$\eta_h = 1 - \frac{0.42}{(\log D_{ref} - 0.172)^2}$$

where D_{ref}=referred impeller diameter, mm.

The following formula is recommended for determining the referred diameter:

$$D_{ref} \approx (4 \text{ to } 4.5)\, 10^3 \sqrt{\frac{Q_s}{n}} \quad \text{mm}$$

where Q_s = discharge, cu m per sec
n = impeller speed, rpm.

Thus, if the capacity, head and power consumption of a pump are known at one speed of the impeller, these parameters can be computed for any other required speed.

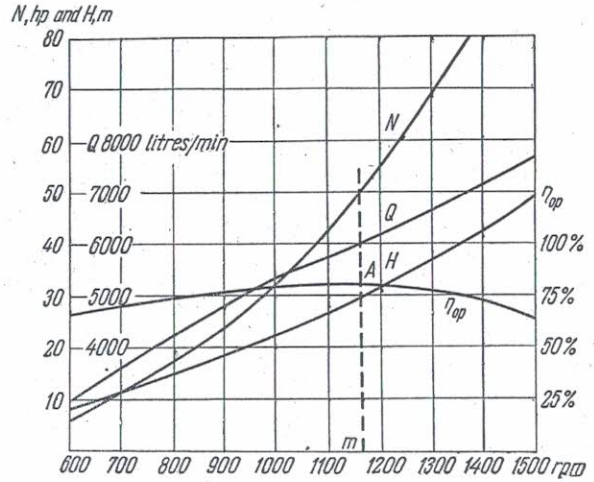


Fig. 123.

Fig. 123 shows the main characteristic curves of a pump and the variation in its efficiency. The line mA indicates the parameters that correspond to maximum efficiency.

If necessary, the head of a pump can be reduced (not more than 15 per cent) by turning down the outside diameter of the impeller. In this case, the new parameters will be

$$Q' = Q\left(\frac{D_2'}{D_2}\right) \quad H' = H\left(\frac{D_2'}{D_2}\right)^2 \text{ and } N' = N\left(\frac{D_2'}{D_2}\right)^3$$

Let us consider the pump characteristics for a constant impeller speed. Here the pump capacity will be changed by regulating the opening of the discharge valve, i.e., by changing the characteristics of the discharge pipeline.

We shall use the following expression to find the mathematic law for the variation of the main theoretical characteristic, as the functional relationship between the head and the capacity is called:

$$H_{t\infty} = \frac{u_2^2}{g} + \frac{u_2 c_{2r}}{g} \cot \beta_2^\circ$$

Since $c_{2r} = \frac{Q_{ts\infty}}{\pi D_2 b_2}$, if the thickness of the vanes is neglected, then

$$H_{t\infty} = \frac{u_2^2}{g} + \frac{u_2}{\pi D_2 b_2 g} Q_{ts\infty} \cot \beta_2^\circ \qquad (258)$$

Equation (258) will be of the following forms for one impeller of a multistage pump with i impellers and for multi-impeller pumps with k_i impellers operating in parallel:

$$\left.\begin{aligned} H_{t\infty} &= \left(\frac{u_2^2}{g} + \frac{u_2}{\pi D_2 b_2 g} Q_{ts\infty} \cot \beta_2^\circ\right)\frac{1}{i} \\ H_{t\infty} &= \frac{u_2^2}{g} + \frac{u_2}{\pi D_2 b_2 g}\frac{Q_{ts\infty}}{k_i} \cot \beta_2^\circ \end{aligned}\right\} \qquad (259)$$

According to equation (259) the nature of the variation in the theoretical head developed by the pump depends upon the outlet angle, β_2°, of the relative velocity from the vanes.* If fact, upon an increase in $Q_{ts\infty}$:

if $\beta_2^\circ > 90°$ and $\cot \beta_2^\circ < 0$, then the theoretical head is decreased,
if $\beta_2^\circ = 90°$ and $\cot \beta_2^\circ = 0$, then the theoretical head remains constant,
if $\beta_2^\circ < 90°$ and $\cot \beta_2^\circ > 0$, then the theoretical head is increased.

These variations of the theoretical head are represented in Fig. 124 by the straight lines *1*, *2* and *3*.

When $Q_{ts}=0$, i.e., when the discharge valve is closed, the head developed by the vanes, and independent of their curvature, is called the initial or shut-off head:

$$H_{t\infty v} = \frac{u_2^2}{g}$$

All pumps in operation deliver liquid through a pipeline. Thus, if we superimpose the characteristic curves of the pipeline and the pump, the intersection of the curves will be the working point of the system.

In Fig. 124 the characteristic curve H_{pl} of the pipeline is superimposed on the curves $H_{t\infty}$ of pumps with various angles β_2°, and the

* Angle β_2 is also called the vane adjustment angle at impeller outlet.

working points are A, B and C. These points correspond to capacities of Q_A, Q_B and Q_C and heads represented by the lines Q_AA, Q_BB and Q_CC.

The working points of a system enable an estimation to be made of the required overall size and rotational speed of the pump impeller.

Let us assume that it is necessary to select a pump for working point B. Passing through point B are the characteristic curves *4*, *5* and *2* of pumps with various types of vanes, which develop the heads

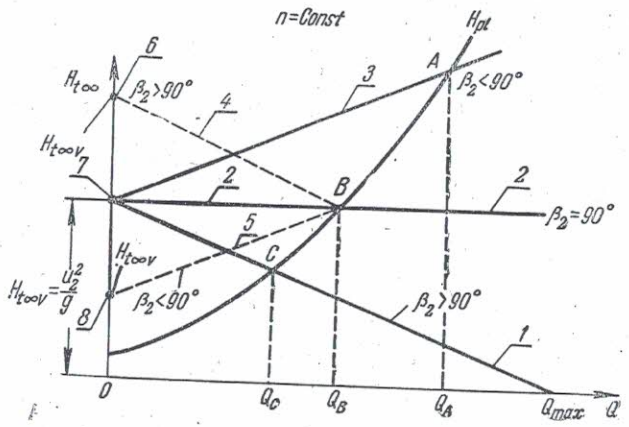


Fig. 124.

6, *7* and *8* when the discharge valve is closed. If these pumps have the same impeller speed, the pump with the backward-curved vanes will be of the largest overall size since its head (point *6*) will be the highest when the discharge is closed. On the other hand, the pump with the forward-curved vanes will be of the smallest overall size since its shut-off head (point *8*) will be the lowest.

If, however, these pumps have impellers of the same diameter, then the pump with the backward-curved vanes will have the highest impeller speed and that with the forward-curved vanes, the lowest rotational speed.

Pumps do not always operate under the most expedient conditions for which they have been designed. It may be necessary for them to operate with either underload or overload. This changes the nature of the liquid flow in the impeller passages which, in turn, influences the head and capacity of the pump and, consequently, its characteristic curves. Under normal load the impeller passages are filled with liquid; when the pump is underloaded, the liquid separates from the

rear walls of the vanes and when it is overloaded, from the front walls.

Let us plot a characteristic curve in which the finite number of vanes and the pump losses have been taken into account.

The variation in the theoretical head will conform to the equation

$$H_t = H_{t\infty} k_{2cu} \left(\frac{u_2^2}{g} + \frac{u_2}{\pi D_2 b_2 g} Q_{ts} \cot \beta_2^\circ \right) \tag{260}$$

Denoting the loss of head in the pump as h_{hn}, theoretical normal head as H_{tn} and the designed actual head under normal operating conditions as H_n we can write

$$\eta_h = \frac{H_n}{H_n + h_{hn}} = \frac{H_{tn} - h_{hn}}{H_{tn}}$$

then

$$h_{hn} = (1 - \eta_h) H_{tn} \tag{261}$$

Since, as a first approximation, the losses in head can be taken proportional to the square of the discharge, the lost head when the discharge is changed from Q_{ns} (normal discharge) to Q_s, in litres per second, can be determined (without the shock losses) from the equation

$$h_h = (1 - \eta_h) H_{tn} \left(\frac{Q_s}{Q_{ns}} \right)^2 \tag{262}$$

To these losses must be added the losses in head from shock (water hammer) due to the separation of the liquid from the walls, eddy formation and the change in velocity in the impeller passages. The latter are

$$h_{sh} = 5 - 0.061 Q_s^2$$

These losses can be computed approximately from the formula

$$h_{sh} = \frac{k_{sh}}{2g} \left[u_1^2 + \left(u_2 \frac{H_t}{H_{t\infty}} \frac{D_2}{D_3} \right)^2 \right] \left(1 - \frac{Q}{Q_n} \right)^2$$

$$h_{sh} = \frac{k_{sh}}{2g} \left[u_1^2 + \left(u_2 k_{2cu} \frac{D_2}{D_3} \right)^2 \right] \left(1 - \frac{Q}{Q_n} \right)^2 \tag{263}$$

where k_{sh} = 0.6 to 0.8 = experimental factor taking into account the shock losses upon inlet onto the vanes

k_{2cu} = circulation factor which can be taken constant as a first approximation (see Table 30).

The approximate value of the head with the discharge valve closed is

$$H_{dv} = \frac{k_{dv} u_2^2}{2g}$$

where k_{dv} = a coefficient (see Fig. 125*a*).

A design working characteristic curve is plotted in Fig. 125*b* in accordance with the preceding formulas. Here

$$H = H_{t\infty} - H_{2cu} - h_h - h_{sh} \qquad (264)$$

After computing H_t and $H_{t\infty}$ for the corresponding discharges, a curve showing the variation of the hydraulic efficiency can be plotted.

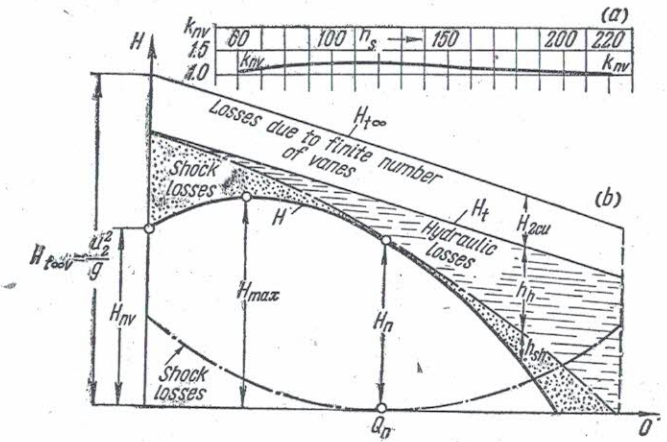


Fig. 125.

Fig. 126 shows actual working characteristic curves which were plotted from values of H and N measured in pump tests for selected values of Q. The efficiency curve was plotted by points computed from the formula

$$\eta_{op} = \frac{\gamma Q_s H}{75N}$$

Universal characteristic curves, a combination of the main and working characteristics, are of great significance in determining the pump parameters for various operating conditions.

Universal characteristic curves are plotted as follows.

Let us assume that head measurements H_1, H_2 and H_3 have been made for the pump speeds n, $0.9n$ and $0.8n$, and that the pump efficiency curves η_{1op}, η_{2op} and η_{3op} have been obtained for the corresponding capacities.

We draw a horizontal line a_1b_1 whose intersections G, F, D, C, B and A with the curves η_{op} are definite values of pump efficiency. These values are projected upwards to intersect the corresponding characteristic curve H. Thus, we find points G_1, F_1, D_1, C_1, B_1 and

A_1, all having the same efficiency value $\eta_{op}=0.65$. Joining these points with a smooth curve we obtain a line of equal efficiencies. Continuing in the same manner we can plot lines for other values of the efficiency.

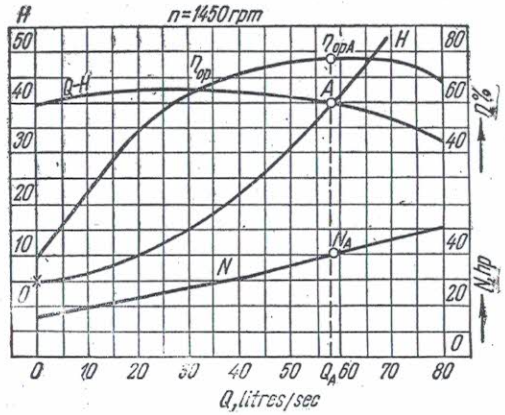


Fig. 126.

Next, the discharge pipeline characteristic curves H_{pl1}, H_{pl2} and H_{pl3} are plotted and the working points K_1, K_2 and A_1 are obtained together with their corresponding parameters.

It is evident from Fig. 127 that the maximum efficiency, $\eta_{op}=0.7$, corresponds to the pipeline characteristic, H_{pl2}.

(B) REGULATION IN PUMP OPERATION

In practice it is often necessary to alter the performance of a pump by regulating the discharge and head.

Pump regulation methods may be classified as either quantitative or qualitative.

Quantitative regulation is usually resorted to on constant-speed pumps and is effected by:

(1) adjusting the valve in the discharge line,

(2) adjusting the valve in the suction line,

(3) passing liquid from the discharge line back to the suction line.

Qualitative regulation is accomplished by varying the rotational speed of the pump impeller; the resulting variations in the pump parameters have been described in the proceeding section.

The discharge is regulated by the quantitative method using the inlet valve (in the suction line) in cases when it is necessary to

maintain a constant pressure at the pump outlet and there is no danger of cavitation occurring at impeller inlet.

Though the quantitative method of discharge regulation by throttling the discharge valve inevitably wastes energy, it finds wide

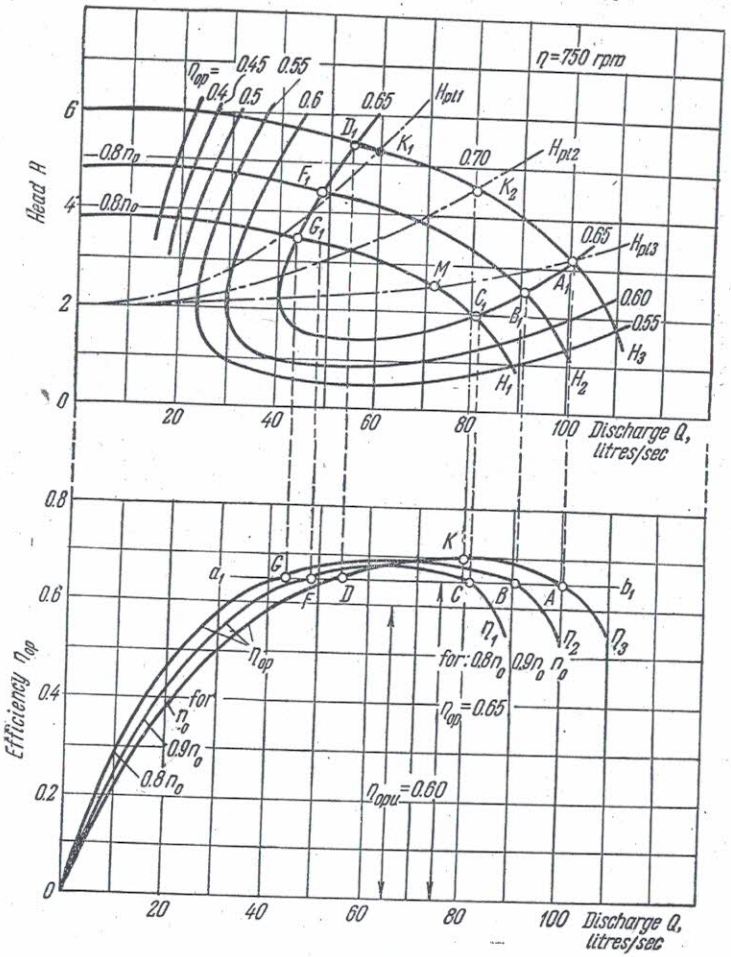


Fig. 127.

application because of its simplicity and can be employed if there is no objection to lowering the pressure in the discharge line.

The method of passing liquid from the discharge line back into the suction line is used when it is undesirable to lower the pressure in the inlet chamber of the pump impeller.

The qualitative method of regulation involves the use of variable-speed motive units or using a hydraulic transmission in place of the coupling. A hydraulic transmission, or fluid drive, enables infinitely variable pump speeds to be obtained with a constant-speed motive unit.

The qualitative regulation method is the most economical but requires more complex equipment.

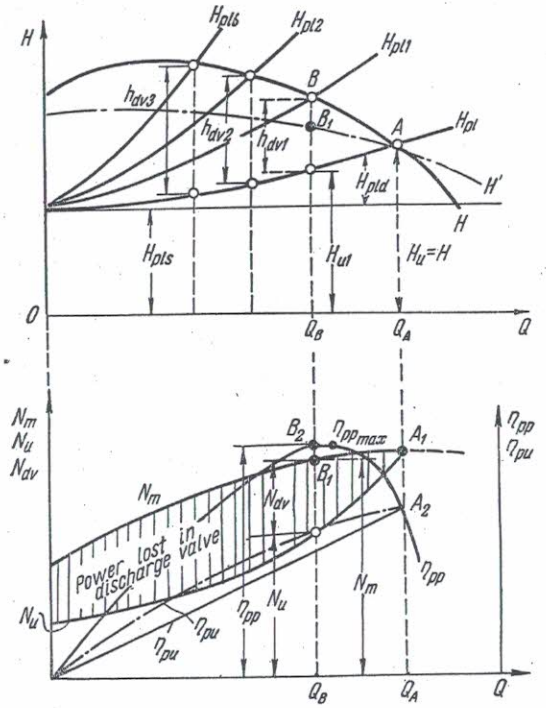


Fig. 128.

We shall consider the quantitative method of pump regulation by throttling the discharge valve, since it is most widely used.

Let us suppose that when the discharge valve is fully opened, the maximum working point is A (Fig. 128) which corresponds to a discharge Q_A and a useful head H_u with a pipeline characteristic H_{pl}.

Closing the discharge valve we change the pipeline characteristic so that it passes through point B of curve H_{pl1} which corresponds to a head H_B and a discharge Q_B.

Now the useful head will be H_{u1} and a part of the head ($H_B - H_{u1}=h_{dv1}$) will be lost in passing through the partly closed discharge

valve. The more the valve is throttled, the steeper the pipeline characteristic curves H_{pl2} and H_{pl3} and the higher the losses h_{dv2} and h_{dv3} will be in the valve. The effective head factor is defined as the ratio of the useful head developed by the pump to the total head:*

$$\eta_{eh}=\frac{H_u}{H} \tag{265}$$

If, when the valve is fully open, the effective head factor $\eta_{eh}=1$ (since $H_u=H$), then η_{eh} will gradually fall as the valve is throttled.

The flatter the characteristic curves of the pump (H') and the pipeline, the higher the effective head factor will be.

The loss of power, transmitted to the pump and dissipated in throttling the liquid in the discharge valve, is determined by the pumping plant utilization factor, η_{pu}, which is the ratio of the useful power, N_u, to the power, N_m, of the motive unit.

If η_{op}, η_m and η_{pp} are the efficiencies of the pump, motive unit and pumping plant, respectively, then

$$\eta_{pu}=\frac{N_u}{N_m}=\frac{\frac{\gamma QH_u}{3{,}600\times 75}}{\frac{\gamma QH}{3{,}600\times 75\eta_m\eta_{op}}}=\frac{H_u}{H}\,\eta_m\eta_{op}=\eta_{eh}\eta_m\eta_{op}=\eta_{eh}\eta_{pp} \tag{266}$$

It is evident from the pumping plant utilization curves in Fig. 128 that the maximum working point does not coincide with the maximum efficiency, $\eta_{pp\,max}$, of the pumping plant, and that the maximum value of $\eta_{p\,u}$ coincides neither with the maximum value of η_{pp} nor with the efficiency when the discharge valve is fully opened. The power lost in throttling the valve is represented by the hatched area. The latter may reach 60 to 70 per cent or more of the power of the pumping plant.

The economy of pumping plant utilization is evaluated by the specific energy, which is the amount of energy, N_s, required to displace 1 cu m per hour of the liquid being pumped.

If N_m is the transmitted power, N_u is the useful power, N_{dv} is the power loss in the discharge valve and Q is the discharge in unit time corresponding to these power values, then

$N_{ms}=\frac{N_m}{Q}$ hp per cu m=specific power transmitted to the pump

$N_{us}=\frac{N_u}{Q}$ hp per cu m=specific useful power

$N_{dvs}=\frac{N_{dv}}{Q}$ hp per cu m=specific power lost in the discharge valve

at most economic operation of the pump.

* The effective head factor for point B is $\eta_{eh}=\frac{H_{u1}}{H}$.

With an increase in the discharge, Q, the specific transmitted power falls and the specific useful power increases, while pumping plant efficiency, η_{pp}, begins to fall after it reaches a certain maximum. It follows that $\eta_{pp\,max}$ and $\eta_{pu\,max}$ do not correspond to the most expedient conditions of pumping plant operation, as is evident in Fig. 129.

In order to ensure pump operation with the required parameters it is necessary to know how the pump characteristics vary with the

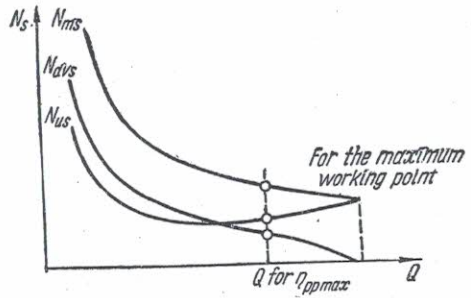


Fig. 129.

degree of discharge valve throttling. We shall now proceed with the plotting of throttling characteristic curves used for this purpose.

The head developed by the pump beyond the discharge valve, in the case of variable discharge and constant impeller speed, is

$$H_{th}=H-h_{dv}=H-\zeta_{dv}\frac{Q_s^2}{2gf_{pl}^2} \tag{267}$$

where h_{dv}=loss in head when the liquid is throttled in the discharge valve

f_{pl}=clear section of the pipeline beyond the valve.

When the valve is fully open, $\zeta_{dv}=0$ and therefore the throttling characteristic coincides with the pump characteristic.

As the valve is gradually closed the value of ζ_{dv} increases so that if we compute the loss values, h_{dv}, for given discharges and degrees of throttling, we can plot a family of throttling characteristic curves using equation (267).

Fig. 130 shows the throttling characteristic curves, H_{th0}, H_{th1}, H_{th2} and H_{th3}, of a pump for four degrees of throttling. The working points will be A, B_1, C_1 and D_1 with the corresponding discharges, Q_A, Q_B, Q_C and Q_D. The useful head is equal to the ordinate between the pipeline characteristic curve and the axis of abscissas. The

head lost in the discharge valve corresponds to the vertical line between the characteristic curve, H_{th0}, for a fully open valve and the curve H_{pt}.

The previously mentioned qualitative method of regulation is based on the variation of pump characteristics at variable rotational speeds of the impeller.

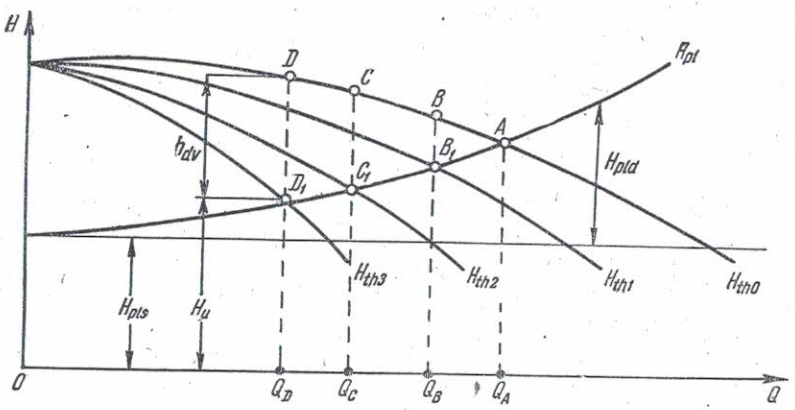


Fig. 130.

If a pump is driven by a variable-speed motive unit its operation can be regulated by a combination of throttling and speed variation. Still another method is to throttle the discharge valve, use a bypass valve and vary the speed of the motive unit.

(C) PUMP OPERATION IN A CIRCUIT

1. Operation with Negative Lift and Unstable Operation

Certain marine pumps operate with negative lift. This must be taken into consideration when the characteristic curves of the system are plotted.

Let us assume that a pump with a characteristic curve H operates with a negative lift, H_{plsn}, and discharges into pipeline *2* (Fig. 131).

First we shall plot the curve for the loss of head, H_{pld1}, in the suction line. If a plane passing through the axis of the inlet hole in the suction line is taken as the datum, the negative lift will be represented by a straight line whose ordinates are H_{plsn}=const. Since a part of the negative lift, H_{pld1}, is lost in overcoming the hydraulic resistances of the suction line, computed values of H_{pld1} (for example, $\overline{ab}$) are laid off downwards from the line H_{plsn} to obtain the negative lift at which the liquid enters the pump. The negative lift will be equal to the ordinates of the curve H_{pld1}.

The static head of pipeline *2* (Fig. 131), into which the pump discharges, is equal to H_{pls2} and is represented by the line MB. If we lay off the losses in head, H_{pld2} (for example, $\overline{cd}$), in overcoming the hydraulic resistances in pipeline *2* upwards from line MB, we shall obtain the curve H_{pld2} which is the characteristic curve of this pipeline. A part of the head, H_{pls2}, equal to the ordinates of curve H_{pld1} (for example, $\overline{ab}$), is overcome by means of the negative lift. Therefore, the pump must overcome the static head, $\Delta H_{pls}=H_{pls2}-(H_{plsn}-H_{pld1})$. Taking the axis MB as the base of the ordinates of the characteristics of the system, we find that the pipeline characteristic curve H_{pl} can be plotted by laying off from the curve H_{pld2} the corresponding lengths, ΔH_{pls}, between the straight line H_{pls2} and curve H_{pld1} (for example, $\overline{bc}=\overline{Ad}$).

Assuming that the pump characteristic curve H is of the shape shown in Fig. 131, it is evident that the maximum working point will be A. The head developed by the pump is H_A and the discharge is Q_A. If the negative lift is increased, characteristic curve H_{pl} will be flatter and point A will be displaced to the right. Thus, the head developed by the pump will decrease and the discharge will increase.

Now let us assume that a different pump with a characteristic curve H_3 is connected to the pipeline. It discharges into pipeline *3* and tank *14*. As the tank is filled and its liquid level rises, the characteristic curve of the pipeline changes consecutively from H_{pl3} to H_{pl7} and the discharge is gradually decreased. At the pipeline characteristic H_{pl7} a failure in delivery will occur and the head developed by the pump will drop to H_{cv}, the value for a closed valve. When this happens the liquid will begin to flow back through the pump to the place from where it was initially delivered.

This reversed flow continues until the pressure in the system falls to the value H_{pl3} after which the pump will again deliver liquid into the tank as before. The failure in delivery will be repeated when the level in the tank reaches a value corresponding to the characteristic curve H_{pl7} as before. This is known as an unstable characteristic curve and such pump operation is considered unstable.

The performance of pumps operating under unstable conditions, with a characteristic curve that is intersected twice by the pipeline characteristic to obtain two working points (for example, *10-10'*, *11-11'*, etc.), may vary with the load. Moreover, the change in performance and failure in delivery are accompanied by water hammer which is highly undesirable. To ensure normal operation of a pump

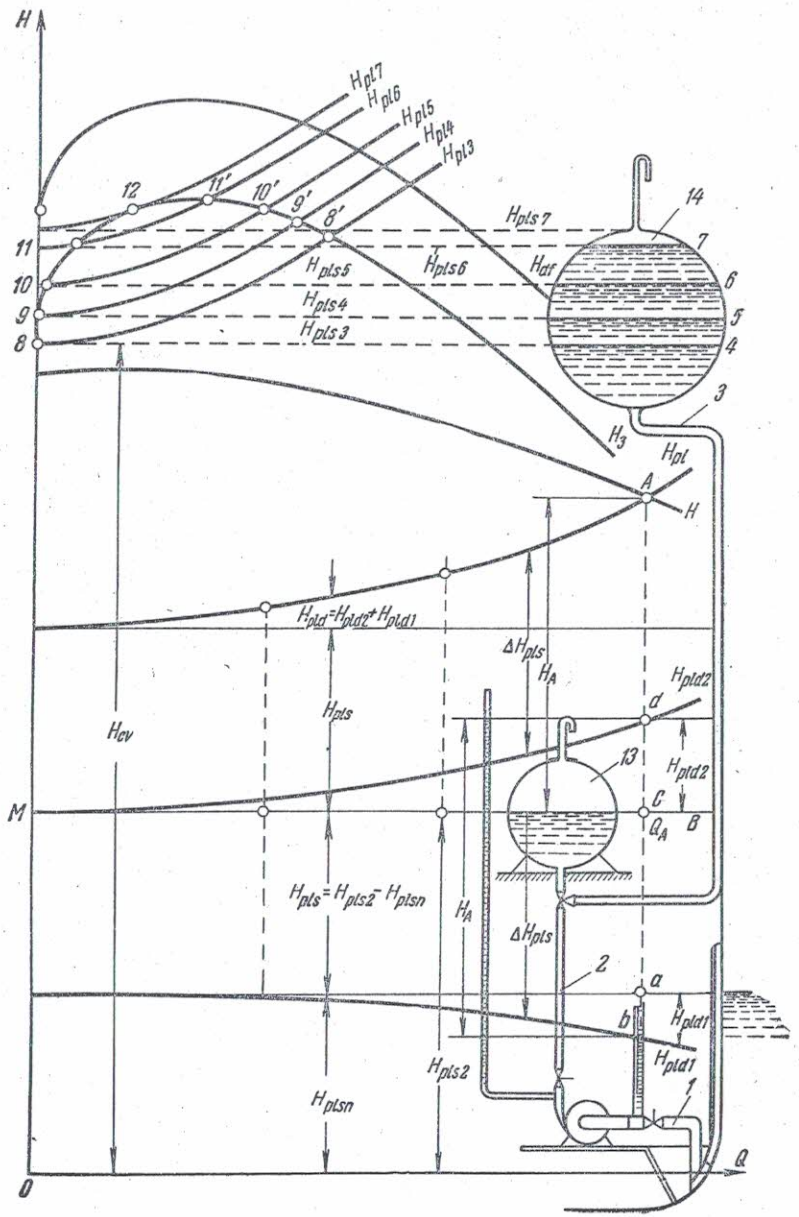


Fig. 131.

with an unstable characteristic curve, H_{df}, the head at the beginning of the delivery failure region must be higher than the manometric head of the system (for example, characteristic curve H_{df}).

2. Pump Operation in a Complex Circuit

Pumps often operate in a complex branched circuit instead of discharging into a simple pipeline.

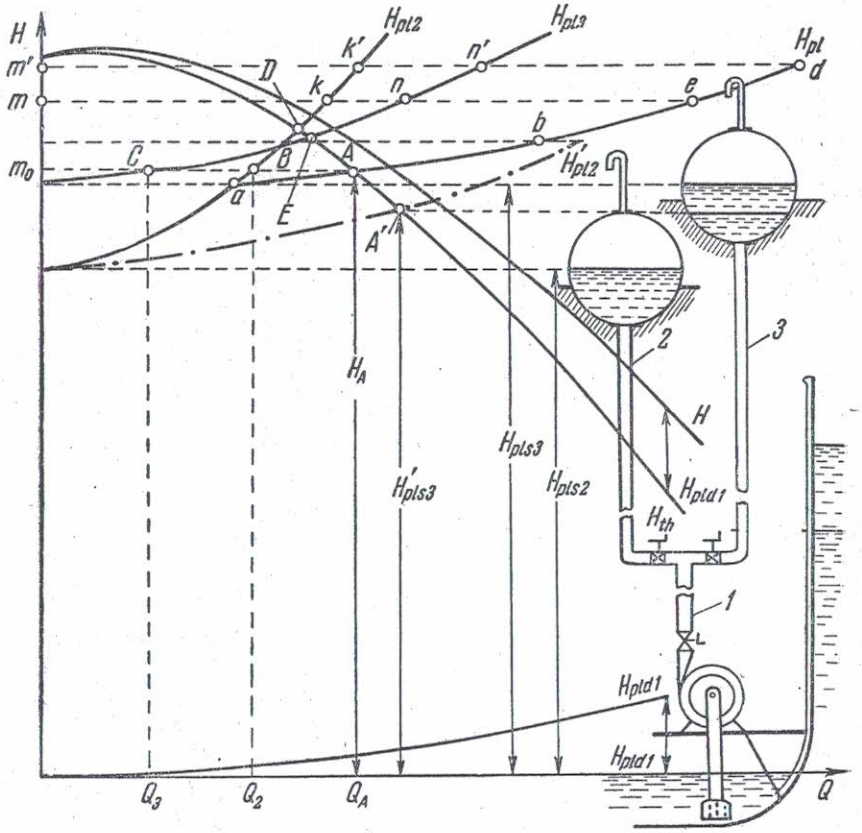


Fig. 132.

Let us consider a case in which a pump operates in the layout shown schematically in Fig. 132. The characteristic curves of the pipelines *1*, *2* and *3*, plotted from calculations made by the methods indicated in the previous sections, are denoted as H_{pld1}, H_{pl2} and H_{pl3}. Only the dynamic part of the characteristic, H_{pld1}, is computed for

pipeline *1* since its static component is included in the characteristics of pipelines *2* and *3*.

Taking into consideration that the useful head is the same in each pipeline connected in parallel, we can plot the summary characteristic curve for the two pipelines.

Adding the abscissas of the curves H_{pl2} and H_{pl3} for points with the same ordinates, i.e., $\overline{m_0B}+\overline{m_0C}=\overline{m_0A}$, $\overline{mn}+\overline{mk}=\overline{me}$ and $\overline{m'n'}+\overline{m'k'}=\overline{m'd}$, we find points A, b, e and d through which a line is drawn from point a to obtain the curve H_{pl}. The latter is the characteristic curve of the system.

Next we plot the throttling characteristic curve, H_{th}, of the pump by subtracting the loss in head, H_{pld1}, required to overcome the hydraulic resistance in pipeline *1* from the ordinates of the given pump characteristic curve H.

The maximum working point of the system is A which determines the pump discharge, Q_A, and head H_A. If the pump discharges separately into pipelines *2* and *3*, the working points will be D and E, respectively.

The discharge through each pipeline is represented by the lengths $\overline{m_0C}=Q_3$ and $\overline{m_0B}=Q_2$ on the line m_0A passsing through the maximum working point A. Since $\overline{m_0C}+\overline{m_0B}=\overline{m_0A}$, then $Q_A=Q_3+Q_2$.

If the throttling characteristic curve, H_{th}, of the pump passes through point a, the pump will discharge only into pipeline *2*. Let us suppose that pipeline *2* has the characteristic curve H'_{pl2}. In this case the level in the tank of pipeline *3* will drop to the height H'_{pls3} since the static head of the pipeline is higher than that developed by the pump which is represented by point A'.

3. *Multiple-Pump Operation in a Circuit*

Sometimes a capacity is required that cannot be provided by any one of the shipboard pumps. In such cases two pumps are connected for parallel operation to deliver liquid into a single system of piping, for instance, into the fire, boiler feed, condensate circulation or drainage systems.

The need may arise under shipboard conditions for heads higher than can be developed by any one of the available pumps. In this case pumps are connected to operate in series.

We shall consider the most frequently encountered cases of multiple-pump operation.

Parallel operation of two identical centrifugal pumps. First we shall take a case in which two pumps (*I* and *II*) with identical characteristic curves H_I are connected in parallel for discharge into a circuit (Fig. 133*a*) having a static head H_{pls} and a summary pipeline charac-

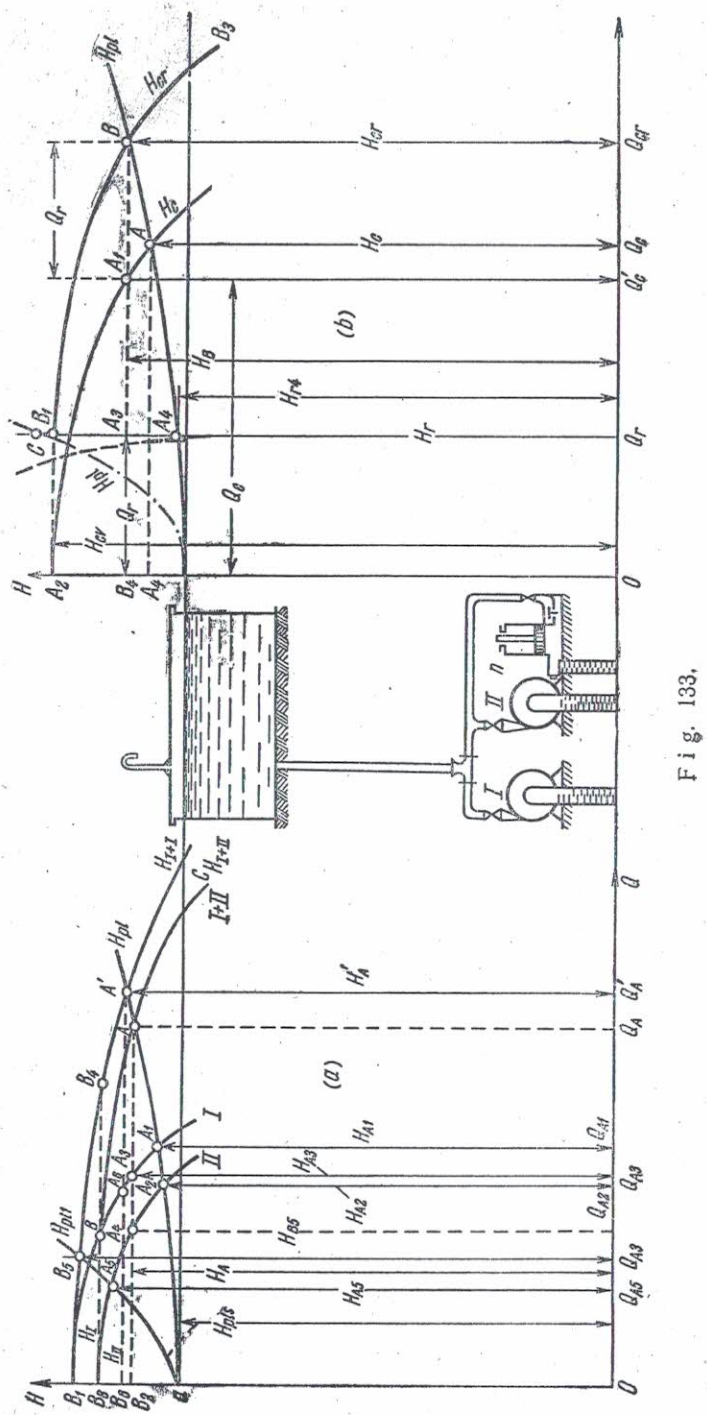


Fig. 133.

teristic curve H_{pl}. Upon discharge of either pump separately into the pipeline with the characteristic H_{pl} the maximum working point is A_1 which corresponds to the discharge Q_{A1} and the head H_{A1}.

The summary characteristic curve of the two pumps when they discharge into one pipeline is represented by the curve H_{I+I}. It is obtained by adding the discharges of the pumps at the same head (for example, $B_3B+B_3B=B_3B_4$). Now the maximum working point is A' corresponding to the discharge $Q_{A'}$, and head $H_{A'}$. It can be seen from the diagram that the summary discharge of the pumps in parallel operation in a single pipeline is less than the sum of their discharges when delivering liquid separately into the same pipeline, i. e., $Q_{A'} < 2Q_{A1}$. This can be explained by the fact that there is a linear relation between the liquid velocity in the pipeline and the discharge while the relation between the velocity and the head is a parabola. The discharge of each pump in parallel operation is equal to one half of the total discharge through the pipeline and is represented by the length $A'A_6=A_6B_6$ while the head is the ordinate OB_6. The steeper the characteristic curve of the pipeline, the less the increase in discharge achieved by parallel operation of two pumps (for instance, for characteristic curve H_{pl1}). Parallel operation of pumps with a steeply drooping characteristic curve will provide for a larger increase in discharge. The discharge can be doubled for a given head by a corresponding reduction of the resistances in the pipeline which may even involve an increase in its diameter.

Parallel operation of two different centrifugal pumps. If the two pumps *I* and *II* have different characteristic curves, H_I and H_{II}, and they discharge separately into a pipeline with a characteristic curve H_{pl} (Fig. 133*a*), their maximum working points will be A_1 and A_2. These points correspond to the discharges Q_{A1} and Q_{A2}, and the heads H_{A1} and H_{A2}. In parallel operation of these pumps the summary characteristic *I* will be represented by curve B_1BC and the working point will be *A* with the discharge Q_A and the head H_A. The summary discharge $Q_A < Q_{A1}+Q_{A2}$. The share of each pump in the summary discharge of both pumps is found on line AB_2 as the length $AA_3=Q_{A3}$ for pump *II* and the length $A_3B_2=AA_4=Q_{A4}$ for pump *I*.

If alterations in the pipeline change its characteristic curve to H_{pl1}, it will be of no avail to connect pump *II* to the circuit since the loss in head in the pipeline will be higher than the head developed by this pump. If, however, it is connected to the circuit, part of the liquid discharged by pump *I* flows back through the passages of pump *II*, leading to water hammer in this pump and reducing the delivery to the pipeline.

Parallel operation of a centrifugal and a reciprocating pump. A reciprocating pump has a flat characteristic curve as do all rotary pumps. Thus, due to its positive action the discharge does not depend upon the head the pump develops. Theoretically, the characteristic of a reciprocating pump is represented by the vertical line H_r (Fig. 133*b*); actually the characteristic deviates somewhat to the left since the discharge is reduced due to leakage at higher heads.

If a centrifugal pump with a characteristic curve H_c operates in parallel with a reciprocating pump having a characteristic H_r we can plot the summary characteristic curve CB_1B_3 by adding the discharges (abscissas), Q_c and Q_r, at the same heads (ordinates).

When the reciprocating and centrifugal pumps operate separately and discharge into a circuit with a characteristic curve H_{pl} the working points will be A_4 and A, respectively. The reciprocating pump will have a discharge Q_r and develop a head H_{r4}; the corresponding values will be Q_c and H_c for the centrifugal pump.

In parallel operation of the two pumps the maximum working point is *B* from which the summary discharge can be determined, as well as the head H_{cr}. The discharge of the centrifugal pump in this case will be $Q_c' < Q_c$ since $A_3B < A_4A$ while the discharge of the reciprocating pump will be $Q_r' = Q_r$ (since $B_4A_3=A_1B$).

Thus, due to the reduced discharge of the centrifugal pump the summary discharge of the two pumps operating in parallel will be $Q_{cr} < Q_c + Q_r$.

If the characteristic of the pipeline is changed to H_{pl1} it will be useless to connect the centrifugal pump since the head it develops is less than the loss of head in the pipeline (point *C*).

Series operation of centrifugal pumps. If identical pumps *I* and *II* with a characteristic curve $H_{I\,(II)}$ operate individually and discharge into the tank *3* (Fig. 134*a*) they can only deliver liquid to the height H_{pls1}. To fill the tank to the height H_{pls2} the pumps are connected by pipe *4* to operate in series. The new pipeline characteristic will be H_{pl2}. In this case the summary characteristic curve is plotted by doubling the head values (ordinates) for the same discharges (abscissas) of the curve $H_{I\,(II)}$.

Drawing a smooth curve through the points plotted in this manner we shall obtain the characteristic H_{I+II} for the operation of pumps *I* and *II* in series.

The working point will be A_1 when each pump discharges separately into a pipeline with a characteristic curve H_{pl1}, and *C* when the pumps operate in series. These points correspond to discharges Q_{A1} and Q_C, and heads H_{A1} and H_C. According to the diagram the discharge of the pumps operating in series will be higher than that of a single pump discharging into a pipeline with a characteristic curve H_{pl1}. As the liquid level in tank *3* rises, point *C*, moving to the left and upwards along the characteristic curve H_{I+II}, will reach its maximum position *A* when the static head of the pipeline is H_{pls2}.

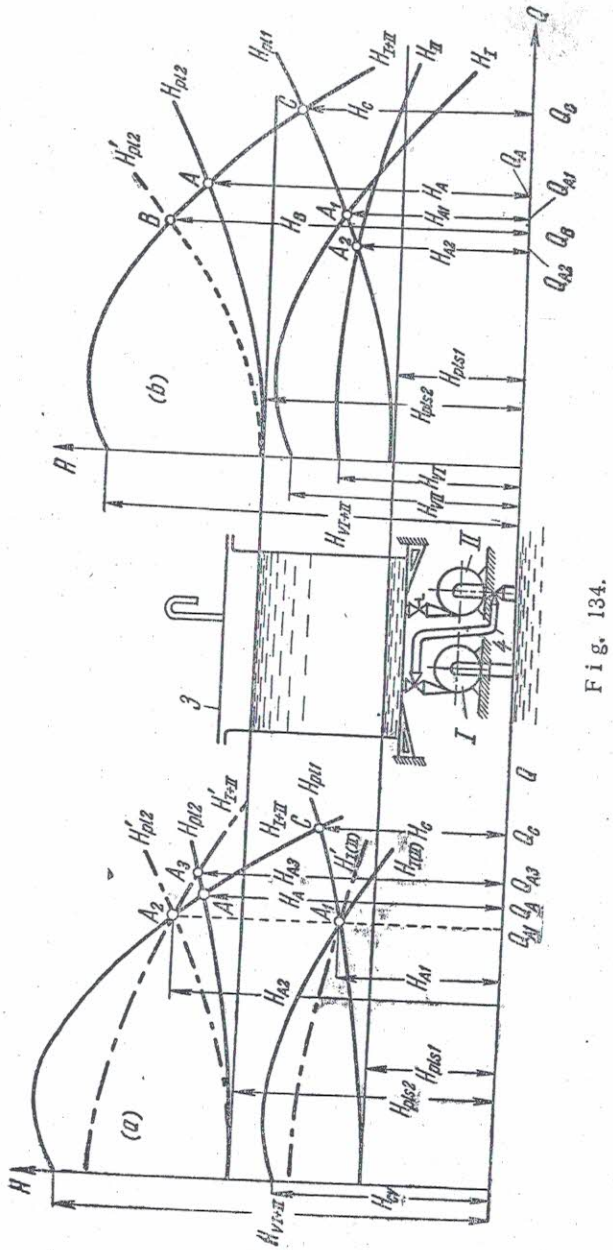


Fig. 134.

Point A corresponds to a discharge $Q_A > Q_{A1}$ and a head $H_A > H_{A1}$.

If the pumps have flat characteristic curves, such as $H'_{I\,(II)}$, then the summary characteristic curve for their operation in series is represented by the curve H'_{I+II}. Here the maximum working point will be A_3 which corresponds to a discharge $Q_{A3} > Q_A$ and head $H_{A3} > H_A$.

Therefore, the flatter the characteristic curves of the pumps, the more their summary discharge will increase when they are connected for series operation.

The discharge of pumps operating in series can be regulated by throttling the valve in the discharge line as shown in Fig. 134*a* by point A_2.

In cases when the pumps I and II have different characteristic curves, H_I and H_{II} (Fig. 134*b*), the working points will be A_1 and A_2 if the pumps operate separately, discharging into a pipeline with the characteristic curve H_{pl1}. The corresponding discharges will be Q_{A1} and Q_{A2}, and heads H_{A1} and H_{A2}. The summary characteristic curve for operation of the pumps in series is plotted from points obtained by adding the heads (ordinates) of curves H_I and H_{II} at the same discharges (abscissas). The new curve H_{I+II} will have the working point C.

The discharge, Q_C, and head, H_C, at point C are higher than the heads and discharges when each pump discharges separately into tank *3*. As the level rises in the tank, point C moves upwards along curve H_{I+II} and, at the level H_{pls2}, it reaches point A. At this point the discharge $Q_A > Q_{A1} > Q_{A2}$ and the head $H_A > H_{A1} > H_{A2}$.

The hydraulic resistance of the pipeline can be varied by throttling. This regulates the discharge of the system (point B).

Pumps can operate in series in a circuit only if the total hydraulic resistance of the pipeline H_{pl} is less than their summary head, H_{vI+II}, when the discharge valve is fully closed.

(D) ADVANTAGES AND DISADVANTAGES OF CENTRIFUGAL PUMPS, AND THEIR APPLICATIONS

Centrifugal pumps are employed both for high and low discharges and heads. The efficiency of properly manufactured centrifugal pumps operating at high capacities reaches that of reciprocating pumps. At low and medium capacities the efficiency of a reciprocating pump is higher than that of a centrifugal pump.

Disadvantages of the centrifugal pump include:

1. The functional dependence of the head on discharge makes it difficult to regulate pump delivery and does not permit the head to be substantially increased whenever necessary if the speed of the motive unit is limited.

2. Since these pumps cannot operate with dry suction it is necessary before starting to prime the inlet chambers and suction line by means of an auxiliary device. This complicates operation and does not allow the pump to be put into action immediately when required. The application of a vacuum device complicates the pumping installation and increases the initial cost.

3. Centrifugal pumps have low efficiencies at low load.

4. The efficiency drops considerably upon an increase in the viscosity of the liquid being pumped.

5. It is difficult to design low-capacity centrifugal pumps that can operate against a high head.

6. Any leakage in the suction line is more detrimental to the operation of a centrifugal pump than to that of a reciprocating pump.

7. The efficiency of the pump depends upon its duty.

Favourable features of centrifugal pumps are:

1. Uniform delivery and constant pressure at steady operation.

2. Small overall size and low weight in comparison with reciprocating pumps.

3. High rotational speed, enabling high-speed motive units to be employed without the use of reducing gears. Some centrifugal pumps operate at speeds as high as 10,000 rpm.

4. Simplicity of construction.

5. Dependability in operation.

6. Low susceptibility to clogging by foreign matter in the liquid being handled.

7. Less current repairs because of the small number of moving parts.

8. Simple regulation of discharge.

These advantageous features have found centrifugal pumps wide application in ships as pumping mechanisms. High-capacity non-priming centrifugal pumps are used in negative-lift pumping plants. Self-priming centrifugal pumps are used in installations with positive lift where it is not necessary for the liquid to be delivered immediately after the pump is started.

Low-speed single-stage volute-casing centrifugal pumps with a capacity up to 400 tons per hour are used as liquid-transfer, drainage, ballast, sanitary, circulation (for internal combustion engines) and oil pumps. In addition, they are used as oil-transfer pumps with capacities up to 200 tons per hour, circulation pumps of condensator installations with capacities up to 6,500 tons per hour, condensate and boiler-feed pumps with capacities up to 500 tons per hour and fire pumps with capacities up to 250 tons per hour.

Multistage diffuser-ring pumps are used for boiler-feed and fire duties. They are of the same capacities as above but operate against a higher head.

The characteristics and overall sizes of these pumps are listed in the corresponding reference manuals and catalogues.

(E) STARTING AND STOPPING CENTRIFUGAL PUMPS, ROUTINE ATTENTION AND POSSIBLE FAULTS IN OPERATION

Economic and reliable operation of pumps, as well as the wear of their parts, depend to a great extent upon careful observance of instructions and pump operating rules. Special attention must be paid to discovering and eliminating the causes of abnormal operation.

Before starting a centrifugal pump it is necessary to make an external inspection, make sure that lubricant is normally supplied to the friction parts, rotate the impeller by hand several turns to check whether any foreign objects impede pump operation, open the vent cocks and the inlet connection valve, fill the suction line and pump casing with the liquid to be pumped, open the cocks of the vacuum and pressure gauges, and open the cocks supplying the liquid to the hydraulic seals of the stuffing boxes. Priming is stopped when the liquid begins to come out of the air vents. This indicates that air has been eliminated from the suction line and casing. Make sure that the liquid is not lost through leaks in the suction line. Pumps equipped with a device for creating a vacuum in the suction line require no priming. When, after starting the motive unit, it reaches its normal speed and the pressure gauge indicates a pressure corresponding to the idle run value (see § 1-30, A), open the valve in the discharge line. Power consumption in idle running equals from 0.35 to 0.5 of the normal power of the motive unit.

The pump can be operated only several minutes with the discharge valve closed. Prolonged operation leads to overheating of the parts and the pump may jam since all the power of the motive unit transmitted to the impeller is converted into thermal energy.

Routine attention during operation involves: (1) watching the instruments to see whether the pump is operating normally; (2) checking whether the liquid is being supplied to the hydraulic seals; (3) checking lubricant feed to the lubricating stations and (4) bleeding air periodically from the casing by opening the air vents.

The pump is stopped by stopping the motive unit, closing the valves in the discharge and suction lines, and closing the cocks of the vacuum and pressure gauges and of the hydraulic seals.

Troubles that appear in pump operation may be due to design, erection and/or operating features of the installation.

Failure of the pump to deliver liquid may be due to one of the following causes:

(1) the pump is installed too high above the level of the liquid being pumped;
(2) the discharge valve is closed;
(3) the pump casing and suction line were not primed before starting;
(4) the liquid is frozen in the inlet line and valves;
(5) the inlet strainers and filters are clogged;
(6) the foot valve leaks when closed;
(7) the hydraulic resistance has increased in the discharge and suction lines (see § 1-21);
(8) leakage in the joints of the suction line, casing and/or stuffing boxes results in air inflow in the suction line;*
(9) the temperature of the liquid being handled is too high;
(10) the valves of the pump are not fully open;
(11) the rotational speed of the impeller is too low;
(12) the inlet hole of the suction line is too near to the bottom of the tank;
(13) the impeller rotates in the wrong direction due to a mistake in installation;
(14) the pump characteristics do not suit the characteristics of the pipeline;
(15) the static suction lift does not suit the cavitation properties of the pump.

Further reasons why a pump does not operate at full capacity, or the discharge gradually drops to zero (delivery failure), may be:
(1) the impeller vanes are damaged;
(2) the pump shaft is running at excessive speed leading to cavitation and failures in delivery;
(3) the free surface of the liquid at the inlet strainer is so low that the pump draws air together with the liquid;
(4) the pump impeller or impellers have shifted axially;
(5) the speed of the motive unit shaft has dropped;
(6) the inlet valve is not fully open.

Overloading in the motive unit of the pump may be due to:
(1) clogging of the pump passages or the casing;
(2) jamming of the shaft journals in the bearings, or other mechanical faults;
(3) large axial shift of the shaft due to excessive wear of the thrust bearings so that the impeller interferes with the internal surfaces of the casing;
(4) excessive axial thrust;
(5) the density and viscosity of the liquid being pumped are much higher than those for which the pump was designed.

* The leak can be found by the deviation of a candle flame with which likely places are tested.

If the pump does not develop the required head, the speed of the impeller is too low or the impeller and shaft packing device are excessively worn.

Excessive vibration of the pump cassing may be the result of:
(1) loose foundation bolts;
(2) misalignment of the motive unit and pump shafts in excess of the permissible limits;
(3) excessive clearance in the drive coupling;
(4) insufficient rigidity of the foundation;
(5) a poorly balanced impeller;
(6) sag in the pumping plant shafts;
(7) cocking of the rotary parts due to loosened fastening devices;
(8) interference between the impeller and casing due to excessive wear of the bearings and sag of the shaft;
(9) the rotational speed approaching the critical value;
(10) foreign objects getting into the pump passages;
(11) operation with cavitation.

The pump casing may be heated owing to:
(1) prolonged operation with the discharge valve closed;
(2) pumping liquid with an excessively high temperature.

The following faults may lead to overheating of the pump shaft:
(1) excessive or nonuniform compression of the packing;
(2) insufficient clearance between the shaft and packing gland;
(3) misalignment of the pumping plant shafts;
(4) cocking of the packing gland;
(5) hardening of the packing.

The most frequently encountered trouble in centrifugal pump operation is overheating of the bearings. It is due to:
(1) excessively tight or worn bearing liners;
(2) a sprung or misaligned pump shaft;
(3) misalignment of the motive unit and pump shafts in excess of the permissible limits;
(4) failure of the lubricating device;
(5) lubricant leaking out of the bearings;
(6) insufficient lubrication;
(7) low quality or polluted lubricant.

All faults in operation must be discovered and eliminated without delay to avoid a breakdown in the pumping installation and abnormal operation of the mechanisms and machines served by the pump.

1-31. Propeller Pumps

A feature common to all propeller (axial-flow) pumps is that the general direction of liquid flow through the pump is purely axial. Their design is based on the aerofoil theory proposed by Professor

N. Zhukovsky. These pumps are relatively small in size and are available for capacities from 0.1 to 30 cu m per sec, operating against heads up to 20 m. On shipboard they may be used as circulating pumps of steam plant condensers, and sometimes as coolant pumps for internal combustion engines, if they are arranged for operation with

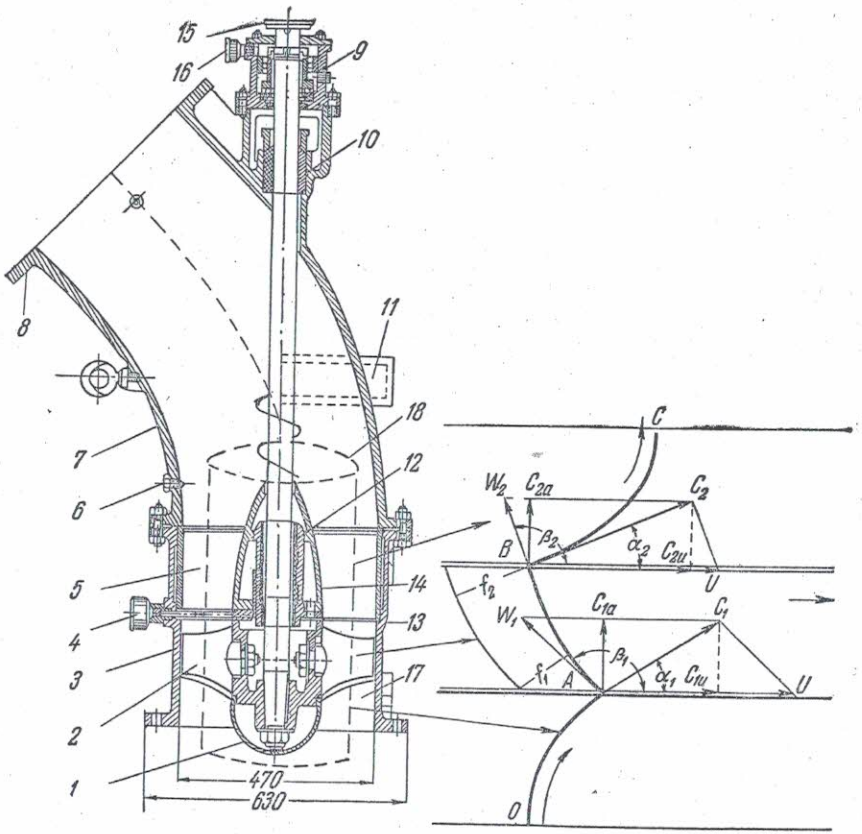


Fig. 135.

negative lift and the required head is low. Propeller pumps may be either of the horizontal- or vertical-shaft type. Either type may have a single impeller or several linked together consecutively.

One design of propeller pump is illustrated in Fig. 135.

The pump casing, consisting of the discharge connection *7* and inlet connection *3*, contains the impeller *2* which is keyed on the pump shaft. The latter is linked to the motive unit shaft by

coupling *15*. The upper end of the pump shaft runs in a radial ball and a ball thrust bearing mounted in housing *9*. The lower end runs in plain bearings *12* and *10*.

Bearing *12* is secured in the cowl *14* which is cast integral with the fixed guide vanes *5*. The guide vane housing is clamped between the shoulders of connections *3* and *7*. The bearings are lubricated through grease cups *16* and *4*. Sometimes the liner of bearing *12* is made of rubber and lubricated by water. The pump is fastened to the hull frame by lugs *11*. The hole for connecting the pressure gauge tube is closed by plug *6*.

The end of the propeller hub enters a fairing *1* cast integral with the inlet guide vanes *17* and connection *3*. This arrangement is designed to impose true axial flow on the liquid, i. e., to remove any whirl components before the liquid enters the rotating vanes. The liquid being pumped enters the impeller *2* through inlet connection *3*. In its motion through the passages between the impeller vanes, the liquid participates in straight flow along the axis of the pump shaft and in a rotary motion about this axis.

Thus, the liquid flows along a helical surface when it is passing through the impeller. The torque of the motive unit is transmitted to the liquid by the rotating vanes thereby increasing its energy. The liquid is discharged from the impeller into guide vanes *5* where recuperation (conversion of dynamic into static head) takes place due to the reduction in the velocity of the liquid flow. Moreover, the guide vanes impart an axial direction to the liquid velocity. Liquid is delivered through flange *8* along the discharge line to its destination. Impellers are usually designed with three, four or five vanes.

We shall now proceed to determine the head, generated by a propeller pump, by developing the cross section of the vanes cut by cylinder *18* of any convenient radius, using the notation shown in Fig. 135.

According to Euler's equation for the theoretical head developed by an impeller, we can write

$$H_{t\infty} = \frac{u_2^2 - u_1^2}{2g} + \frac{w_1^2 - w_2^2}{2g} + \frac{c_2^2 - c_1^2}{2g} \qquad (268)$$

Assuming that the liquid in the passages of the impeller flows parallel to its axis, thus implying that the peripheral velocities at entry and exit from the vanes are equal ($u_1 = u_2$), we obtain

$$H_{t\infty} = \frac{w_1^2 - w_2^2}{2g} + \frac{c_2^2 - c_1^2}{2g} \qquad (269)$$

It follows from equation (269) that the head developed by the pump is the result of the conversion of dynamic into static head as

the liquid flows through the pump passages. Therefore, the clear cross-sectional area of the impeller passages $f_1 < f_2$. The theoretical head generated by the pump can also be expressed by the formula

$$H_{t\infty} = \frac{u_2 c_{2u} \cos \alpha_2^\circ - u_1 c_1 \cos \alpha_1^\circ}{g} = \frac{u}{g}(c_{2u} - c_{1u}) \text{ mH}_2\text{O} \quad (270)$$

Next we shall find the relation between the angles β_1° and β_2° of the liquid velocities on the impeller vanes.

In accordance with the velocity diagram (Fig. 135), and after the necessary transformation, we can write

$$c_{2u} = u_2 - c_{2a} \cot (180^\circ - \beta_2^\circ)$$

$$c_{1u} = u_1 - c_{1a} \cot (180^\circ - \beta_1^\circ)$$

$$c_{2u} - c_{1u} = [u_2 - c_{2a} \cot (180^\circ - \beta_2^\circ) - u_1 + c_{1a} \cot (180^\circ - \beta_1^\circ)]$$

As a rule, propeller pumps are designed so that

$$c_{1a} = c_{2a} = c_a$$

then

$$c_{2u} - c_{1u} = c_a [\cot (180^\circ - \beta_1^\circ) - \cot (180^\circ - \beta_2^\circ)] = c_a (\cot \beta_2^\circ - \cot \beta_1^\circ) \quad (271)$$

Combining equations (270) and (271)

$$H_{t\infty} = \frac{u}{g} c_a (\cot \beta_2^\circ - \cot \beta_1^\circ) \quad (272)$$

Since the value $H_{t\infty}$ is always greater than zero we can write the following inequalities

$$\cot \beta_2^\circ > \beta_1^\circ$$

and

$$\beta_2^\circ < \beta_1^\circ$$

Thus, the impeller of a propeller pump must have forward curved vanes.

It is evident from equation (272) that when the difference $\cot \beta_2^\circ - \cot \beta_1^\circ$ is reduced the theoretical head drops and reaches zero when $\beta_1^\circ = \beta_2^\circ$.

Academician G. Proskura's formula can be used for a tentative calculation of the actual head developed by a propeller pump:

$$H = \frac{u^2}{2gk_u^2} \text{ m}$$

where $k_u^2 = 0.0244\, n_{s1}^{2/3}$ = head factor for a specific velocity $n_{s1} = 500$ to 2,000.

Let us use the following notation:

n = 600 to 2,000 = rotational speed of the pump impeller, rpm

Q_s = discharge of the pump, cu m per sec

$D_i = (4.5 \text{ to } 5.5) \sqrt[3]{\frac{Q_s}{n}}$ = diameter of the impeller, m

D_{hub} = diameter of the impeller hub, m

$m = \frac{D_i}{D_{hub}} = 2$ to 3.5 = ratio of the impeller to hub diameter (the higher the speed the higher m will be)

$k_c = 0.0055\, n_{s1}^{2/3}$ = axial velocity factor

$c_a = c_1 = k_c \sqrt{2gH} = 5$ to 7 = axial velocity of flow in the impeller passages, m per sec.

Hence the pump capacity will be

$$Q = 3{,}600\, \eta_v c_a \frac{\pi}{4}(D_i^2 - D_{hub}^2) = 2{,}826 \eta_v (m^2 - 1) D_{hub}^2 c_a = 2{,}826\, \eta_v (m^2 - 1) D_{hub}^2 k_c \sqrt{2gH} \text{ cu m per hour}$$

where $\eta_v \approx 0.9$ to 0.95 = volumetric efficiency.

Propeller pumps cannot operate with dry suction and are usually arranged for negative lift.

The maximum suction is determined by the formula

$$z_s = \frac{p_a}{\gamma} - \frac{p_v}{\gamma} - \sigma H - h_l \text{ mH}_2\text{O}$$

where $\sigma = \left(\frac{n_{s1}}{560}\right)^{4/3}$ = cavitation factor

h_l = loss in head in the suction line, mH_2O.

The power required to drive a propeller pump is

$$N = \frac{Q_s \gamma H}{75 \eta_{op}} \text{ metric hp}$$

where $\eta_{op} = 0.75$ to 0.9 = overall efficiency of the pump.

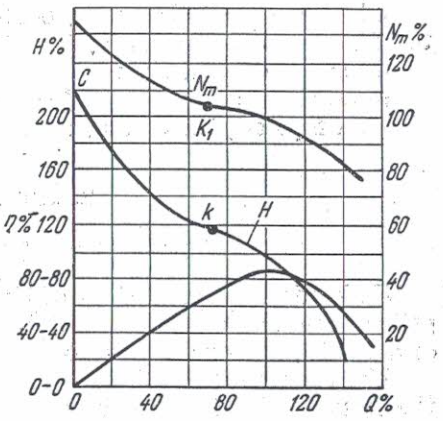


Fig. 136.

Characteristic curves of a propeller pump are shown in Fig. 136.

The power vs discharge N_m and the head vs discharge H curves both have points of inflection, K_1 and k. The power consumption is 20 to 40 per cent higher and the head 50 to 100 per cent higher when the discharge valve is closed than these values at maximum efficiency.

It proves inexpedient to regulate the discharge by throttling with this type of characteristic curve since this leads to a considerable drop in efficiency.

For this reason, the discharge is regulated in high-capacity pumps by varying the pitch of the vanes, i.e., their angle in reference to the plane of the impeller.

Another feature of variable-pitch pumps is that the vanes can be turned into a plane for starting. In this case, the liquid will offer only low resistance to rotation thereby reducing the required starting torque of the motive unit.

Pump operation on the sections of the characteristic curves before the inflection points (left section) is accompanied by water hammer and therefore avoided.

The power consumption and head developed by the pump decrease with an increase in the discharge. This explains, to some extent, why these pumps are widely employed for high capacities.

The head can be increased by having several impellers operating in series.

The main disadvantage of propeller pumps is in the low head they can develop. Their advantages include:

(1) high efficiency;
(2) high rotational speed, enabling them to be coupled directly to high-speed motive units;
(3) small overall size;
(4) ability to pump liquid containing foreign matter;
(5) simplicity in design, maintenance and operation.

E. JET PUMPS

1-32. General Considerations

Under definite conditions of liquid flow through jet pumps, their principle is based on the possibility of converting the static energy of a liquid, gas or steam into kinetic energy and vice versa.

The vacuum in the inlet chamber of a jet pump, required to draw the liquid being handled into the chamber, is created by the unidirectional flow of a full stream of the actuating fluid.

According to the physical state of the actuating fluid, jet pumps are classified as:

(1) water-jet, in which water is employed as the actuating fluid;
(2) steam-jet, in which steam is the actuating fluid;
(3) gas-jet, in which the conditions required for pump operation are provided by a gas (usually air).

Depending upon the way in which the pumps are connected in the installation, they are classified as:

(1) suction, or extraction, pumps, where the pump inlet is connected to the object being served;
(2) delivery pumps, where the pump outlet is connected to the object being served;
(3) vacuum pumps, where the inlet is connected to the object in which a vacuum is created.

According to their purpose, jet pumps are classified as:

(1) discharge pumps designed to remove liquid from the object;
(2) feed pumps delivering feed water into boilers;
(3) air pumps used for removing the air from condensers of steam power plants and the suction lines of centrifugal pump installations;
(4) ash pumps for disposing of cinders and ashes from the boiler and engine rooms;
(5) fish pumps used for conveying the catch from fishing boats to the shore.

Steam-jet pumps which can develop a pressure higher than that of the live steam are called injectors and are employed as auxiliary boiler-feed facilities. All other types of jet pumps are called ejectors.

Ash and fish pumps are of the water-jet ejector type.

Condenser installations of steam power plants are usually served by steam-jet vacuum ejectors in conjunction with condensate pumps. Both steam and water is used as the actuating fluid in discharge ejectors and air vacuum ejectors of pump installations.

1-33. Water-Jet Ejectors

The application of water-jet ejectors for shipboard duty is due to their many advantageous features. Among these are:

(1) the absence of moving parts, as a result of which these pumps have a long service life;
(2) their silent and dependable operation;
(3) their ability to pump liquids heavily polluted with foreign matter;
(4) the impossibility of clogging such pumps;
(5) their ability to operate with dry suction;
(6) rapid starting and suction of the liquid;
(7) uniform delivery;
(8) their ability to create a vacuum and develop the required head;
(9) small overall size and weight;
(10) ease of manufacture;
(11) their ability to operate immersed in the liquid.

The disadvantages of water-jet pumps are their low efficiency and the fact that they are not self-contained.

A typical water-jet pump is shown schematically in Fig. 137. Here, pump *5* delivers the actuating fluid (water), drawn in through

connection *9*, into the converging conical nozzle *1* at a pressure head H_p mH_2O. As the stream of water flows out of nozzle *1* into the mixing chamber *2*, pressure head is converted into dynamic head. This leads to a drop in pressure in the chamber and a vacuum is created at nozzle outlet.

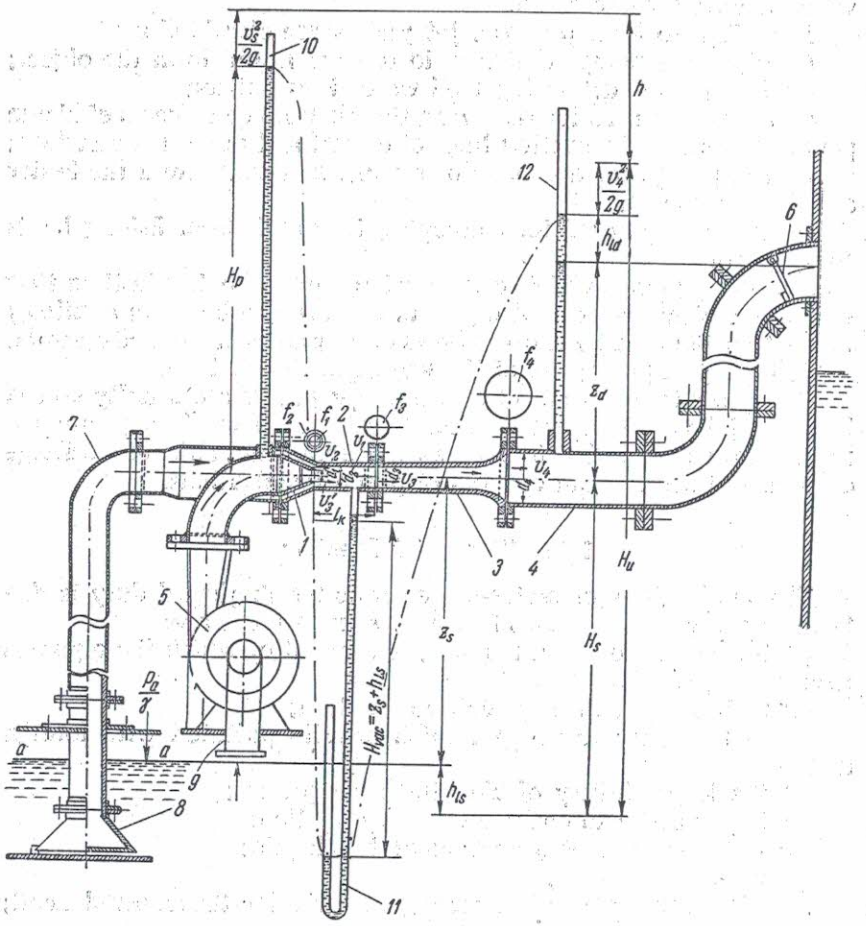


Fig. 137.

When the pump is started, mixing chamber *2* will be full of air. The jet of water discharged from the nozzle *1* at a high velocity entrains a part of the air in the mixing chamber due to the friction between them resulting from viscosity. From mixing chamber *2* the

mixture of water and air passes into the gradually diverging diffuser *3* in which its velocity is reduced to that in the discharge line, and in which recuperation takes place. The mixture enters the discharge line *4* with a higher static head and, passing through flap valve *6*, is discharged overboard. As the air is drawn out of the suction pipe *7*, the vacuum in the pipe increases. The head, $\frac{p_a}{\gamma}$, of atmospheric air acting on the free surface, a-a, of the liquid forces the liquid through the inlet connection *8* and upwards in the suction pipe.

When the liquid being pumped reaches the height z_s and enters the mixing chamber *2*, it is delivered further through the diffuser *3*, discharge line *4*, flap valve *6* and overboard. At this point, the pump begins steady operation.

At steady operation the vacuum gauge *11* indicates a vacuum H_{vac} which is equal to the sum of the static, z_s, and so-called dynamic, h_{ls}, losses in the suction line, while the piezometer *10* indicates the pressure head H_p of the actuating fluid. The reading of piezometer *12* will equal the static z_d and dynamic h_{ld} losses in the discharge line.

If an actuating fluid consumption of Q_{s1} cu m per sec provides for a liquid discharge of Q_{s2} cu m per sec, the useful power, N_u, applied power, N, and pump efficiency, η_{op}, will be

$$N_u = \gamma_2 Q_{s2} H_u$$

$$N = \gamma_1 Q_{s1} (H_u + h)$$

$$\eta_{op} = \frac{N_u}{N} = \frac{\gamma_2 Q_{s2} H_u}{\gamma_1 Q_{s1}(H_u + h)} = \beta_e \frac{H_u}{H_u + h} \approx 0.15 \text{ to } 0.3$$

where H_u = useful head, mH_2O

h = loss of head in the jet pump, mH_2O

γ_1 and γ_2 = densities of the actuating fluid and the liquid being pumped

$\beta_e = \frac{Q_{s2}\gamma_2}{Q_{s1}\gamma_1} = 1$ to 1.5 = ejection factor.

In addition to stationary water-jet pumps, a portable design may be used on ships to drain compartments not accommodated by inlet branches of the drainage system.

In this case, the actuating water is supplied to the pump from the fire mains.

A water-jet ash pump is shown in Fig. 138. Actuating water is delivered from the fire mains at a pressure of 16 to 20 atm (abs) to valve *6*. From here it passes through nozzle *11* located in throat *8* of casing *1* and creates a vacuum in the throat. When cover *7* is open, ashes and refuse are thrown on grate *10* from where they drop into the casing. The jet of water flowing out of the nozzle at a high

velocity carries the ashes and refuse up through pipe *2* and valve *3* and discharges overboard.

Since the internal surface of cover *9* is subjected to intensive wear it has a replaceable deflector *4*.

All water-jet ash-disposal pumps, regardless of their design, are started in the following manner. First the pump supplying the

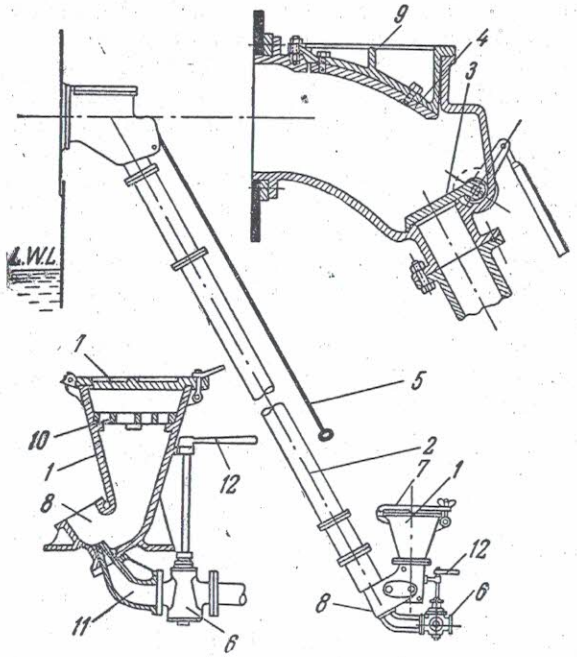


Fig. 138.

actuating fluid to the ejector is started. When the pump develops the required working head, stop valve *6* is opened rapidly with handle *12*. Injector cover *7* is still closed at this time. If the ejector does not "deliver", it is necessary to close valve *6* rapidly and then open it again.

When the ejector begins to operate, cover *7* is opened and ashes, refuse and cinders are shovelled onto the grate.

The pressure of the actuating fluid must be maintained constant during ejector operation. If the pressure drops or the fluid supply fails, pipe *2* may become clogged with ashes.

To stop the ejector it is necessary to close the cover of the ash bunker, stop the pump delivering the actuating fluid, close valve *6*

on the ejector, drain the water from the pipelines and close the drainage valves.

The penetration of sea-water into the compartment where the ejector is installed, when it is not operating, is prevented by keeping cover *7* closed and by closing valve *3* with rod *5*.

Troubles that occur in ash ejector operation are usually due to clogging of the nozzle, faults in installation and excessive wear of the ejector.

1-34. Injectors

Injectors have found wide application as auxiliary boiler-feed devices because of:

(1) their small overall size;
(2) the absence of moving parts;
(3) their ability to operate with dry suction;
(4) the rapid starting and suction of the liquid;
(5) their uniform delivery;
(6) the simplicity of manufacture and maintenance;
(7) the low losses of thermal energy.

A disadvantage of injectors is the impossibility of regulating their discharge.

Steam produced by the boiler it feeds serves as the actuating fluid in the injector. Injectors may be either of the lifting or delivery types.

Lifting injectors can draw in liquid when they are installed above its free surface. Delivery, or nonlifting, injectors operate with negative lift. A distinguishing feature of lifting injectors is the provision of a starting valve which regulates steam admission into the steam nozzle. Delivery injectors do not have such a valve. Only lifting injectors are employed on sea-going ships.

In injectors of up-to-date design, 1 kg of steam feeds from 14 to 15 kg of water, on an average, into the boiler and heats the water to 40°-50°C. The amount of heat returned in this manner to the boiler is from 80 to 90 per cent of the heat in the steam. Thus, only from 10 to 20 per cent of the steam is lost in the process.

Though many different designs of injectors have been devised they all operate on the same principle. We shall consider the operation of the lifting injector whose design and installation is illustrated in Fig. 139.

This injector has three conical members: steam nozzle *1*, combining, or condensing, tube *2* with the hinged half-cone *17*, and delivery tube *3*. Live steam, entering through pipeline *10* and valve *11*, expands in the steam nozzle *1* where it acquires a high velocity, which may reach 600 to 700 m per sec, and a negative gauge pressure, thereby creating a vacuum in chamber *15*.

The pressure of the atmosphere on the free surface in tank *13* forces the feed-water through valve *14* and pipeline *4* into chamber *15*. From here the water passes through the annular gap between nozzle *1* and tube *2* at a velocity of 10 to 12 m per sec into combining tube *2* where it mixes with the steam, promoting the condensation of the latter. At the same time, the velocity of the water increases. As condensation proceeds and the steam-and-water mixture passes along

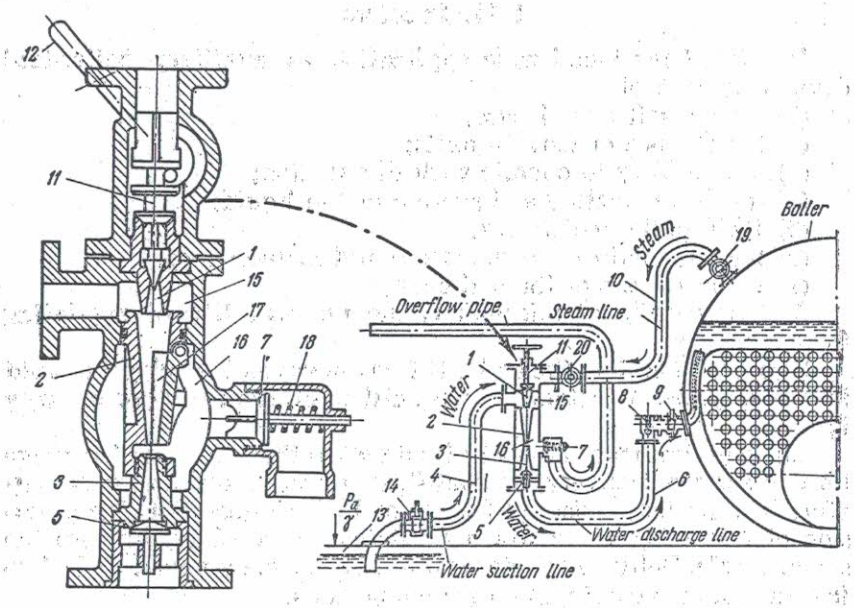


Fig. 139.

the combining tube its volume decreases, this being the reason for the converging shape of the tube. Steam condensation leads to an increase of the vacuum in the chamber of tube *2* thus promoting better suction of the feed-water. The steam-and-water mixture leaves the combining tube *2* at a velocity of 40 to 60 m per sec and enters the mixing, or overflow, chamber *16* equipped with the overflow valve *7*.

The injector is started by turning lever *12*. The action of the head of the water being drawn into the injector and the steam-and-water mixture forces the air from the suction pipe *4* into the overflow chamber *16*, after swinging back half-cone *17*, and then further, through the nonreturn overflow valve *7*, into the atmosphere.

Water following the air and steam-and-water mixture will also flow through the overflow valve indicating that the injector is lifting water. This flow of water through overflow valve *7* continues until the intensity of the action of the steam jet issuing from nozzle *1* begins to exceed the velocity at the orifice of combining tube *2*. This can be accomplished by increasing the steam discharge through the steam nozzle. As soon as the velocity is exceeded, water from overflow chamber *16* is drawn into the stream, pressure in the chamber drops, overflow valve *7* is closed by the pressure of the atmosphere and the action of spring *18*, and half-cone *17* returns to its working position.

As soon as the overflow valve closes, the mixture of feed-water and condensate enters the diverging delivery tube *3*. Here the reduction in velocity raises the pressure of the mixture to a value sufficient to open valve *5* and the nonreturn valve *8* at the boiler. In this manner, feed-water is supplied to the boiler through pipeline *6* when valve *9* is open. The water velocity in the suction and discharge lines is about 1 or 2 m per sec.

Only 0.5 per cent of the mechanical energy of steam flow is imparted to the water in the injector. The other 99.5 per cent is a loss in mechanical energy of the steam.

Valve *11* should be raised slowly with lever *12* in starting the injector. If it is raised too fast a high vacuum will be rapidly created in the suction pipe so that the liquid will boil. The increased pressure will stop water delivery into the injector so that the steam will not be condensed in the combining tube. In this case, steam issues from the overflow pipe through valve *7*.

If the injector draws in water, air will issue first from the overflow pipe followed by the steam-and-water mixture and, finally, water.

The knock heard when the overflow valve drops onto its seat indicates that feed-water is being delivered into the boiler. From this moment valve *11* can be opened completely. The temperature of the feed-water delivered to the injector should not exceed 70°C. At higher temperatures the injector will fail to draw in water.

The maximum discharge of an injector depends upon the clear section of the inlet orifice in the delivery tube. The size designations of injectors are usually numbers equal to the diameter of this orifice in millimetres.

An injector can be started at a boiler steam pressure as low as 1.5 to 2.5 kg per sq cm and it will operate at a pressure of 14 kg per sq cm, a suction lift of up to 5 m and a feed-water temperature of 15°C.

Before starting an injector, valves *19*, *20*, *14* and *9* are opened to admit steam and feed-water to the injector, as well as feed-water to the boiler.

As mentioned above, the injector is started by slowly turning lever *12* of valve *11*. If the injector does not draw in water immediately,

close valve *11* quickly and start again or make any required adjustments. It is best to employ repeated momentary starts to avoid overheating the injector casing, otherwise the injector will fail to deliver.

Normal injector operation depends to a considerable extent upon proper assembly, selection and location of the conical members which should ensure the required ratio between the amounts of steam and water admitted into the injector. The usual cause of abnormal operation of an injector is an incorrect proportion between the steam admitted and the water required for its condensation. Surplus water or steam is discharged through the overflow valve.

The following troubles may be encountered in injector operation:

1. The injector does not deliver water to the boiler but discharges cold water through the overflow valve. Consequently, more water is being admitted to the injector than is required for normal operation. This may be the result of:

(a) too small effective section of the orifice in the steam nozzle;

(b) excessive wear of the internal surfaces in the steam nozzle, and the combining and delivery tubes;

(c) abnormally large annular gap between the steam nozzle and combining tube;

(d) too small annular gap between these members;

(e) the formation of scale in the combining or delivery tubes;

(f) a poor fit of the hinged half-cone to the combining tube;

(g) insufficiently large hole in the inlet part of the delivery tube;

(h) incorrect size of the gap between the combining and delivery tubes;

(i) too small clear passage in the steam valve of the live steam line, or incomplete opening of the valve;

(j) clogging of the steam nozzle.

2. The injector operates but also discharges cold water through the overflow valve. The possible causes are:

(a) too small clear section of the orifice in the steam nozzle;

(b) clogging of the steam nozzle;

(c) large annular gap between the steam nozzle and combining tube.

3. The injector does not draw in water as the result of:

(a) too small effective section of the orifice in the steam nozzle;

(b) clogging of the steam nozzle;

(c) excessive clearance in the fit of the steam nozzle in its seat;

(d) scale in the combining and delivery tubes;

(e) insufficient opening of the overflow valve or its clogging;

(f) leakage in the valves of the steam and feed-water lines, as well as in the injector needle valve, and overheating of the injector casing in starting;

(g) clogging of the valve in the suction line, or incomplete opening; clogging or reduction of the clear section in the suction pipes (denting, flattening, etc.);

(h) excessively high suction lift.

4. The injector operates but discharges hot water through the overflow valve and water delivery fails when the steam valve is opened further. Possible causes are:

(a) the clear section in the steam nozzle is too large;

(b) the annular gap between the steam nozzle and combining tube is too small;

(c) the internal surfaces in the steam nozzle, and the combining and delivery tubes are excessively worn;

(d) scale is formed in the combining or delivery tubes;

(e) size of the hole in the inlet part of the delivery tube is too small;

(f) the head of the combining tube does not fit tightly in its seat so that water leaks from the delivery to the overflow chamber;

(g) water boils in the discharge connection for admitting water into the boiler;

(h) the gap between the combining and delivery tubes is too large.

5. The injector does not operate, failures in delivery are observed in starting and hot water or steam are discharged through the overflow pipe. This may result from:

(a) too large clear section in the steam nozzle;

(b) excessive wear of the internal surface of the steam nozzle;

(c) a loose fit of the head of the combining tube in its seat so that vater leaks from the water to the overflow chamber;

(d) leakage through the overflow valve.

6. The injector discharges cold water through the overflow valve but does not deliver feed-water. The causes may be:

(a) water is delivered to the injector with excessive negative lift;

(b) the valve in the suction line is open too widely;

(c) the valve admitting steam to the injector is not fully open or its clear section is insufficient.

7. The overflow valve vibrates during operation, discharging water at intervals due to:

(a) leakage through the valve;

(b) air inflow in the suction line.

The injector is stopped by turning lever *12* of the steam valve and then closing the stop valves in the steam and boiler-feed lines.

If the injector will not draw water out of the compartments of the double bottom, it should be switched over to supply from a tank whose level is higher than the injector (peak or deep-tank). When the injector begins to operate properly, it is switched back to the double-bottom tank. If the injector is in proper order it will begin to deliver water from this tank.

Chapter 2

FANS

2-1. Required Capacity of Fans and the Head They Develop

Mechanisms designed for delivering gases and increasing their pressure to 1,500 mmH_2O are called *fans*.

According to their principle, fans are classified as centrifugal and axial, and they are driven by steam engine, turbine or electric motor.

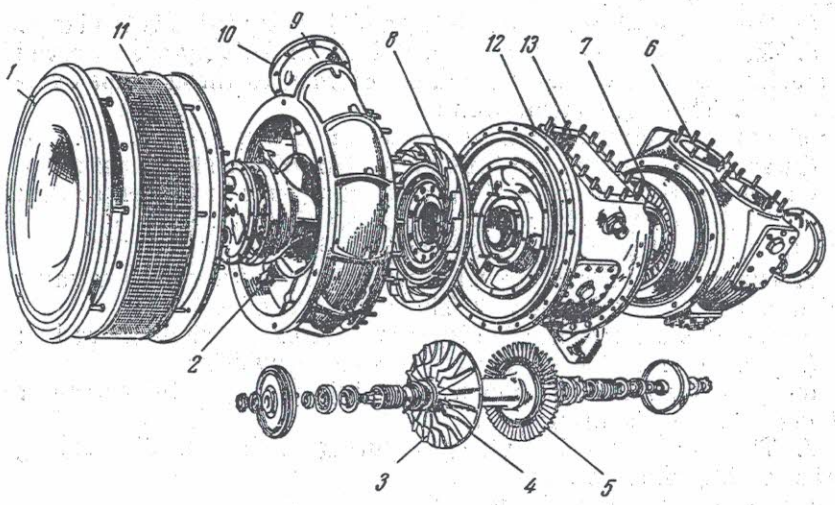


Fig. 140.

A centrifugal fan operates on exactly the same principle as a centrifugal pump.

An exploded view of a typical centrifugal fan driven by a direct-connected steam turbine is shown in Fig. 140. Air enters inlet *1* with screen filter *11* and passes through fixed guide vanes *2* to the entry edges of vanes *3* of the impeller, or wheel, *4*. Upon rotation of impeller *4*, the air, due to the mechanical action of the vanes, acquires a

peripheral velocity of rotation about the impeller axis, a relative velocity along the vanes and an absolute velocity which is the geometrical sum of the peripheral and relative velocities. Energy of the motive unit, transmitted to the fan impeller, is imparted to the air as it flows through the passages between the impeller vanes from the circumference of the inlet edges to that of the outlet edges of the vanes.

The velocity of the air leaving the impeller is considerably higher than that permitted in the air ducts. To reduce its velocity the air

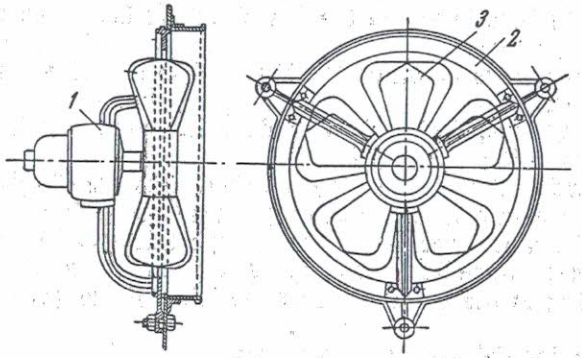


Fig. 141.

passes through the fixed vanes of a diffuser designed on the same principles as that in a centrifugal pump. Thus, the stream of air discharged from impeller *4* enters the fixed vanes of diffuser *8* where the reduction in velocity leads to recuperation (conversion of dynamic into static head). Further reduction of air velocity to that in the air duct takes place in the volute casing *9* where the air acquires the required static head at outlet flange *10*.

The use of two recuperating devices is due to the high velocity of the air leaving the impeller. The latter is driven by a high-speed single-stage Curtis-type turbine wheel *5*.

Wheel *5* is housed in casing *12* into which the nozzle housing *7* enters. The nozzles admit live steam to wheel *5*. Live steam enters the turbine through flanges *6* and is discharged through flange *13*.

The principle of an axial, or propeller, fan in no way differs from that of an axial-flow (propeller) pump.

The propeller fan illustrated in Fig. 141 is of a type used to ventilate shipboard living and service accommodations. It is mounted directly in the bulkhead by means of frame *2*. The propeller *3* of this fan is driven by an electric motor *1*.

According to their purpose fans are classified as:

(1) fans of service and living compart ments, designed to provide induced ventilation in these spaces;

(2) cargo hold fans, designed for ventila ting the holds of dry-store bulk carriers, tankers and refrigerated cargo vessels, as well as refrigerated provision chambers;

(3) boiler plant fans, designed to produce artificial draught for the steam boilers;

(4) coal bunker fans.

Depending upon the way they are installed fans are classified as:

(1) supply fans in which the fan discharge is connected with the spaces being served;

(2) exhaust fans in which the fan inlet is connected to the spaces being served;

(3) ceiling fans, designed to produce air movement in the spaces without providing exchange.

As regards the pressure they develop, fans are divided into:

(1) low-pressure fans developing a head up to 100 mmH_2O;

(2) medium-pressure fans developing a head up to 300 mmH_2O;

(3) high-pressure fans developing a head up to 1,500 mmH_2O.

According to the mechanical composition of the gas they handle, there are:

(1) fans for delivering pure gases;

(2) dust fans designed for delivering gases pollute d by mechanical impurities.

The specific velocity, n_s, of a fan is a value relating the air discharge, Q cu m per hour, full head, H mmH_2O, at normal atmospheric conditions and the fan wheel speed, n rpm, at the highest efficiency:

$$n_s = \frac{n\sqrt{Q}}{\sqrt[4]{H^3}}$$

Gas passing through the fan is compressed to only the slightest degree and is therefore assumed to be an incompressible fluid. In this case all the main principles in the theory and operation of centrifugal pumps are valid for fans as well.

The capacity of a fan for ventilating definite service quarters must be sufficient to maintain the chemical composition, humidity and temperature of the air within the requirements of sanitary regulations.

The unwholesomeness, or contamination, of the air in a room or compartment due to the presence of people is usually estimated by the carbon dioxide content, which increases with an increase of harmful impurities in the air. The carbon dioxide content of the air must not exceed 0.1 to 0.15 per cent by volume.

The fan capacity required to maintain a stipulated chemical composition of the air in a compartment is

$$Q_{ch} = V_r \frac{V_{rc}}{V_{mr} - V_{ca}} \text{ cu m per hour} \qquad (273)$$

where V_{rc} = volume of carbon dioxide produced per cu m of the given room, litres per cu m

V_r = volume of the room, cu m

$V_{mr} \approx 1$ = the maximum carbon dioxide content per cu m of the given room, litres per cu m

$V_{ca} \approx 0.3$ = carbon dioxide content per cu m of sea air entering the room, litres per cu m.

The volume of air required to maintain the prescribed temperature in a room is

$$Q_t = \frac{Q_r}{c_a (t_r - t_{fa})\,\gamma_{fa}} = \frac{Q_r}{c_a (t_r - t_{fa})\,\frac{\gamma_0}{1+\alpha t_r}} = \frac{Q_r (1+\alpha t_r)}{c_a (t_r - t_{fa})}\,\gamma_0 \qquad (274)$$

where $c_a \approx 0.24$ = mean heat capacity of air, kcal per kg °C

t_r = given temperature of the room, °C

t_{fa} = temperature of the fresh air entering the room, °C

Q_r = amount of heat entering the room, kcal per hour

γ_{fa} = density of the fresh air entering the room, kg per cu m

$\gamma_0 \approx 1.29$ = density of dry air at 0°C and a pressure of 760 mmHg, kg per cu m

$\alpha = \frac{1}{273}$ = coefficient of volumetric expansion of air.

The amount of external air required to maintain the relative humidity in a room is

$$Q_{hu} = \frac{100\,D_{hu}}{\varphi_r d_r - \varphi_{fa} d_{fa}} \text{ cu m per hour} \qquad (275)$$

where D_{hu} = amount of moisture entering the room, g per hour

d_r and d_{fa} = absolute humidity of saturated air at the room temperature, t_r, and at the temperature, t_{fa}, of the entering air, g per cu m (see Table 38)

φ_r and φ_{fa} = relative humidity of the air in the room and of the entering air, per cent.

Data on the relative humidity and temperature of the outside air depending upon the locality in which the ship is operating, and the permissible values for various accommodations are listed in Table 39.

The amount of carbon dioxide, heat and vapour produced by persons in a room can be calculated from the data of Table 40.

Table 38

Temperature, °C	Density, kg/m³	Absolute humidity, g/m³	Vapour pressure, mmHg	Temperature, °C	Density, kg/m³	Absolute humidity, g/m³	Vapour pressure, mmHg
—25	1,424	0.64	0,540	+13	1,235	11.32	11.162
—24	1,418	0.71	0,600	+14	1,230	12.03	11.908
—22	1,406	0.86	0,745	+16	1,222	13.59	13.536
—21	1,401	0.95	0,825	+17	1,217	14.43	14.421
—20	1,395	1.05	0,910	+18	1,213	15.31	15.357
—19	1,390	1.15	1,000	+19	1,209	16.25	16.346
—18	1,384	1.25	1,095	+20	1,205	17.22	17.391
—17	1,379	1.35	1,190	+21	1,201	18.25	18.495
—16	1,374	1.46	1,290	+22	1,197	19.33	19.659
—15	1,368	1.58	1,400	+23	1,193	20.48	20.888
—14	1,363	1.70	1,520	+24	1,189	21.68	22.184
—13	1,358	1.83	1,635	+25	1,185	22.93	23.550
—12	1,353	1.98	1,780	+26	1,181	24.24	24.988
—11	1,347	2.14	1,930	+27	1,177	25.64	26.505
—10	1,342	2.31	2,093	+28	1,173	27.09	28.101
—9	1,337	2.49	2,267	+29	1,169	28.62	29.782
—8	1,332	2.69	2,455	+30	1,165	30.21	31.548
—7	1,327	2.90	2,658	+31	1,161	31.89	33.406
—6	1,322	3.13	2,876	+32	1,157	33.64	35.350
—5	1,317	3.37	3,113	+33	1,154	35.48	37.411
—4	1,312	3.64	3,368	+34	1,149	37.40	39.565
—3	1,308	3.92	3,644	+35	1,146	39.41	41.827
—2	1,303	4.22	3,941	+36	1,142	41.51	44.201
—1	1,298	4.55	4,263	+37	1,139	43.71	46.691
0	1,293	4.89	4,600	+38	1,135	46.00	49.302
+1	1,288	5.23	4,940	+39	1,131	48.40	52.039
+2	1,284	5.60	5,302	+40	1,128	50.91	54.906
+3	1,279	5.98	5,687	+41	1,124	53.52	57.910
+4	1,275	6.39	6,097	+42	1,121	56.25	61.055
+5	1,270	6.82	6,534	+43	1,117	59.09	64.346
+6	1,265	7.28	6,998	+44	1,114	62.05	67.790
+7	1,261	7.76	7,492	+45	1,110	65.14	71.391
+8	1,256	8.28	8,017	+46	1,107	68.36	75.158
+9	1,252	8.82	8,574	+47	1,103	71.73	79.093
+10	1,247	9.39	9,165	+48	1,100	75.22	83.204
+11	1,243	10.01	9,792	+49	1,096	78.86	88.499
+12	1,239	10.64	10,457	+50	1,093	82.63	91.982

Table 39

Locality	Data on fresh air and sea water						Air requirements for shipboard accommodations		
	Warmest period of navigation			Coldest period of navigation			Accommodations	Air temperature		Relative humidity φ_r, %
	Temperature of outside air t_{fa}, °C	Water temperature t_w, °C	Relative humidity of outside air φ_{fa}, %	Temperature of outside air t_{fa}, °C	Water temperature t_w, °C	Relative humidity of outside air φ_{fa}, %		Coldest period of navigation, t_r, °C	Warmest period of navigation, t_r, °C	
Rivers that freeze	20 to 30	16 to 25	55 to 65	—5	4	75 to 85	Living and passenger accommodations, state-rooms and ward-rooms	18	t_{fa}+5°C	40 to 60
							Passageways of living and service accommodations	15	t_{fa}+5°C	75 to 85
Seas in high or temperate latitudes	10 to 25	5 to 15	65 to 75	—25 to —35 —15 to —20	0 to —2	80 to 85	Bath- and shower-rooms	25	t_{fa}+10°C	70
							Cloak-rooms	20	t_{fa}+5°C	70 to 80
Warm seas	25 to 30	20 to 25	55 to 65	—20	4	75 to 85	Wash-rooms and laundries	15	t_{fa}+5°C	60
Tropical seas	30	27	70	30	27	70	Toilets	12	t_{fa}+5°C	80
Navigation in any localities	30	27	70	—25	0	80	Galleys	8 to 22	t_{fa}+10°C	80
							Pantries	8	t_{fa}+5°C	80
							Wet provisions and vegetable storage rooms	2	t_{fa}+5°C	80

Table 40

Each adult produces per hour	Carbon dioxide, litres/h	Heat, kcal/h	Vapour, g/h
At hard physical work	45	150	130
At quiet work	23	100	75
At rest	23	75	70
At sleep	23	75	40
Each child up to 12 years of age produces	12	50	23

It should be noted that the amount of vapour produced in a room by the operation of steam engines and steam lines is approximately equal to 1 or 2 per cent of the steam consumption of the engines or lines.

The heat generated by various sources and introduced into the premises they occupy can be determined from the formulas listed in Table 41.

In calculating the fan capacity required for engine and boiler rooms it is necessary to take into consideration the amount of air required for the operation of internal combustion engines and boilers.

The approximate volumes of air required for the operation of internal combustion engines, V_{eic}, and boilers, V_b, are found from the following formulas:

$$V_{eic} = 60\,\alpha_{ex} V_{cyl} n \text{ cu m per hour}$$

where V_{cyl} = total displacement of the cylinders, cu m
n = engine shaft speed, rpm
α_{ex} = 1.3 to 1.5=excess air coefficient.

$$V_b = 1.15\,\alpha_b (1 + \alpha t_{fa})\, B \frac{Q_l}{1{,}000} \text{ cu m per hour}$$

where $\alpha_b \approx 1.2$ to 1.5 = excess air coefficient
α = coefficient of volumetric expansion of air
B = fuel consumption, kg per hour
Q_l = lower calorific value of the fuel, kcal per kg.

The required fan capacities calculated from formulas (273), (274) and (275) will not be the same and therefore the highest value should be taken for any given compartment.

Tentative values of the required capacity can be estimated on the basis of the number of air renewals per hour n_{re} as established by experience for various accommodations (Table 42).

Table 41

Source of heat	Heat emitted into surroundings, Q_i, kcal/h	Notation
Steam boilers	(0.03 to 0.05) $G_f Q_l$	Q_f = total fuel consumption in boiler, kg per hour
Steam turbines	0.005 $G_s \Delta i$	Q_l = lower calorific value of the fuel, kcal per kg
Steam engines	(0.005 to 0.01) $G_s \Delta i$	G_s = steam consumption, kg per hour
Auxiliary machinery	(0.02 to 0.03) $G_s \Delta i$	Δi = useful heat drop, kcal per kg
Steam lines	0.01 $G_s \Delta i$	N_e = effective power
Internal combustion engines	0.02 $N_e g_c Q_l$	N = power, kW
Electrical machinery:		i_e = mean current density per sq mm of conductor cross section, A per sq mm
(a) with recirculating cooling system	64 $N \frac{1-\eta}{\eta}$	η = efficiency of electrical machine
(b) without cooling	864 $N \frac{1-\eta}{\eta}$	g_c = fuel consumption, kg per hp-h
Lighting fixtures	864 N	k = coefficient of heat transmission of room walls, kcal per hour per sq m per deg C
Wires, bus bars, cables and fittings	2,160 i_e^2	F = area of the room walls, sq m
Heat introduced from outside by transmission through ship's hull	$\Sigma kF\,\Delta t$	Δt = difference in temperatures of room wall surfaces and external surfaces of ship's hull, deg C

In this case, if V_{com} is the volume of the compartment in cu m, the required hourly capacity of the fan will be

$$Q_a = n_{re} V_{com} \text{ cu m per hour}$$

The fan capacity needed is selected on the basis of what is called standard air. This means air at a temperature t_{st}=20°C, pressure

Table 42

Compartment	Number of air renewals per hour for	
	Plenum ventilation	Exhaust ventilation
Passengers', officers' and crew accommodations	10 to 15	—
Public rooms (staterooms, dining-saloons, etc.)	15 to 20	10 to 15
Smoking rooms	—	15 to 20
Gymnasiums	15	20
Swimming pools	15	20
Russian baths	—	10 to 20
Galleys	5 to 10	40 to 60
Provision rooms without cooling facilities	5 to 10	10 to 15
Bathrooms, toilets and laundries	5	15 to 20
Sick bays	5 to 10	10 to 20
Baggage rooms	—	20
Deck refreshment bars	10 to 15	25 to 30
Upper deck passageways	—	6
Middle deck passageways	—	7
Lower deck passageways	—	8
Engine and boiler rooms	30	35

$p_{st}=760$ mmHg, relative humidity of $\varphi_{st}=50$ per cent and density $\gamma_{st}=1.2$ kg per cu m. The capacity of the fan determined for air in a given state, having a pressure p_a, volume Q_a and temperature t_a, can be converted to the standard air capacity by using formula (276) which is derived from the equation

$$\frac{p_{st}Q_{st}}{1+\alpha t_{st}}=\frac{p_aQ_a}{1+\alpha t_a}$$

whence

$$Q_{st}=\frac{(1+\alpha t_{st})\,p_aQ_a}{p_{st}(1+\alpha t_a)}=Q_a\frac{\left(1+\frac{1}{273}20\right)}{\left(1+\frac{1}{273}t_a\right)}\frac{p_a}{760}=$$

$$=Q_a\frac{293}{273+t_a}\frac{p_a}{760}\text{ cu m per hour} \qquad (276)$$

The theoretical head developed by the fan is expressed in mm of water column:

$$H_{t\infty}=\frac{1}{g}(c_{2u}u_2-c_{1u}u_1)=\frac{1{,}000}{g}\frac{\gamma_{air}}{\gamma_{wat}}(c_{2u}u_2-c_{1u}u_1)=$$
$$=\rho(c_{2u}u_2-c_{1u}u_1)\text{ mmH}_2\text{O} \qquad (277)$$

where γ_{air} = density of air, kg per cu m
$\gamma_{wat}=1{,}000$ = density of water, kg per cu m
ρ = mass density of air, kg-sec^2 per m^4.

Upon radial entry of the air onto the fan impeller vanes

$$H_{t\infty}=\rho c_{2u}u_2\text{ mmH}_2\text{O}$$

Taking into account the effect of having a finite number of impeller vanes on the developed head by the factor σ and for the losses of head in the fan by the hydraulic efficiency η_h we obtain the actual head

$$H=H_{t\infty}\sigma\eta_h=\sigma\rho c_{2u}u_2\eta_h=\sigma\rho\frac{c_{2u}}{u_2}u_2u_2\eta_h=$$
$$=\sigma\rho\varphi_h u_2^2\eta_h=\rho\psi_h u_2^2\text{ mmH}_2\text{O} \qquad (278)$$

where $\varphi_h=\frac{c_{2u}}{u_2}$ = eddy current factor

$\psi_h=\sigma\varphi_h\eta_h$ = head factor taken equal to: 0.8 to 1.1 for forward-curved vanes; 0.6 to 0.8 for radial, or straight, vanes; 0.5 to 0.7 for backward-curved vanes.

The fan head required to accommodate a given ventilating system depends upon the resistance and characteristic curves of the latter.

The permissible maximum peripheral speeds (tip speeds) of an impeller, based upon fan design and strength considerations, are listed in Table 43. The table also lists the most widely used inlet and outlet angles of the vanes.

Table 43

Type of fan	Peripheral speed, m per sec	Inlet angle	Outlet angle
Low-pressure	30 to 40	95 to 105	15 to 25
Medium-pressure	40 to 50	125 to 130	30 to 35
High-pressure	50 to 90	140 to 145	40 to 45

Backward curved vanes are rarely employed and then only for low-pressure fans. The number of vanes is usually assigned so as to facilitate laying out and may be equal to 4, 6, 8, 12, 16, 24, 32 or 48.

The power required to drive a fan is found from the formula

$$N_m = \frac{Q_a H}{75\eta_f 3{,}600} \text{ hp}$$

The overall efficiency of a fan is made up of the following efficiencies:

1. Hydraulic efficiency, which takes into consideration the loss of head in the fan

$$\eta_h = \frac{H}{H+\Delta H} = \frac{H}{H_t} = 0.7 \text{ to } 0.85$$

where ΔH = loss of head in the fan.

2. Hydraulic friction efficiency which takes into account the losses due to the friction of the impeller shrouds against the fluid being transferred

$$\eta_{fr} = \frac{N_{fr}}{N_a} = \frac{\beta 10^{-6} \rho D_2^2 u_2^3}{N_a}$$

where N_{fr} = power lost in overcoming fluid friction

$\beta = (5 \text{ to } 15)\left(1+5\frac{b_2}{D_2}\right)$ = coefficient obtained from data compiled by the Central Institute of Aero- and Hydrodynamics

b_2 = width of the impeller at air outlet

D_2 = impeller diameter at air outlet

For backward-curved vanes—$\eta_{fr} \cong 0.6$ to 0.75

For forward-curved vanes—$\eta_r \cong 0.75$ to 0.9.

3. Mechanical efficiency which takes into account the losses due to mechanical friction

$$\eta_m = \frac{N_a - \Delta N_{mf}}{N_a} \approx 0.95 \text{ to } 0.99$$

where ΔN_{mf} = power lost in overcoming mechanical friction. The overall efficiency of a fan is thus

$$\eta_f = \eta_h \eta_{fr} \eta_m = 0.4 \text{ to } 0.75 \qquad (279)$$

The overall efficiency of an axial fan may reach $\eta_f \approx 0.84$.

2-2. Design and Selection of Fans

Strictly aerodynamical calculations in fan design do not, as a rule, ensure results in subsequent tests that comply with the initial design data.

More accurate results may be achieved by designing a fan similar to one which has already been built, tested and modified to obtain the most favourable aerodynamic and design features.

This type of fan design is carried out by the similarity method using aerodynamic diagrams and dimensionless characteristics which we will consider in the following.

The initial data for fan design comprise: the total head, H, consisting of the static, H_{st}, and dynamic, H_{dyn}, heads, capacity, Q_g, and the rotational speed, n, at maximum efficiency. Thus

$$H = H_{st} + H_{dyn} = H_{st} + \frac{v^2}{2g} \times 10^{-3} \text{ mmH}_2\text{O} \qquad (280)$$

where v = mean velocity in the discharge connection of the fan. On the basis of the discharge per second, Q_s, head, H, and speed, n, we next determine the specific velocity of the fan.

The specific velocity of a fan is a value that relates the air discharge, Q_{st} cu m per sec, the total head, H mmH$_2$O, and the impeller speed, n, at maximum efficiency:

$$n_s = \frac{n\sqrt{Q_a}}{\sqrt[4]{H^3}} \qquad (281)$$

It is evident that the ratio of the capacities of a series of geometrically similar fans of identical design can be expressed by the dimensionless discharge coefficient $\overline{Q}_k$. Therefore

$$\overline{Q}_k = \frac{Q_s}{Fu_2}$$

and

$$Q_s = \overline{Q}_k F u_2 = \overline{Q}_k \frac{\pi D_2^2}{4} u_2 \text{ cu m per sec}$$

where F = area of the impeller, sq m

D_2 = outside diameter of the impeller, m.

The peripheral speed at the outlet circumference of the impeller is found from the formula

$$u_2 = \frac{\pi D_2 n}{60} \text{ m per sec}$$

The pressure developed by a series of geometrically similar fans can be characterized by the pressure coefficient, $\overline{H}_k$:

$$\overline{H}_k = \frac{H}{u_2^2 \rho} \text{ — for the total head, and}$$

$$\overline{H}_{kst} = \frac{H_{st}}{u_2^2 \rho} \text{ — for the static head.}$$

Whence, if we know $\overline{H}_k$ from the characteristics of pilot models, we can determine

$$H = \overline{H}_k u_2^2 \rho \text{ mmH}_2\text{O}$$
$$H_{st} = \overline{H}_{kst} u_2^2 \rho \text{ mmH}_2\text{O} \qquad (282)$$

We shall call the ratio of the actually consumed power to the theoretical possible power consumption the consumed power coefficient $\overline{N}_k$. Then

$$\overline{N}_k = \frac{75N_m}{\rho F u_2^3}$$

therefore, the actual power applied to the fan shaft is

$$N_a = \overline{N}_k \frac{Fu_2^3\rho}{75} \text{ metric hp} \qquad (283)$$

The overall efficiency of the fan can be determined from the preceding coefficients:

$$\eta_f = \frac{\overline{Q}_k\overline{H}_k}{\overline{N}_k} = \frac{Q_sH}{75N_a} \text{ — for the total head, and}$$

$$\eta_{fst} = \frac{\overline{Q}_k\overline{H}_{kst}}{\overline{N}_k} = \frac{Q_sH_{st}}{75N_a} \text{ — for the static head.}$$

The following order may be followed in fan design with dimensionless coefficients:

1. Determine the specific velocity for the given working conditions:

$$n_s = \frac{n\sqrt{Q_g}}{\sqrt[4]{H^3}}$$

2. On the basis of the obtained value of n_s select a fan with characteristic curves and aerodynamic diagram of the type shown in Fig. 142.

3. At maximum efficiency η_f on the dimensionless characteristic curve for the total pressure find the corresponding pressure coefficient $\overline{H}_k$.

4. Determine the peripheral speed from formula (282):

$$u_2 = \sqrt{\frac{H}{\rho\overline{H}_k}} \text{ m per sec}$$

5. Now we can calculate the outside diameter of the impeller:

$$D_2 = \frac{60u_2}{\pi n} \text{ m}$$

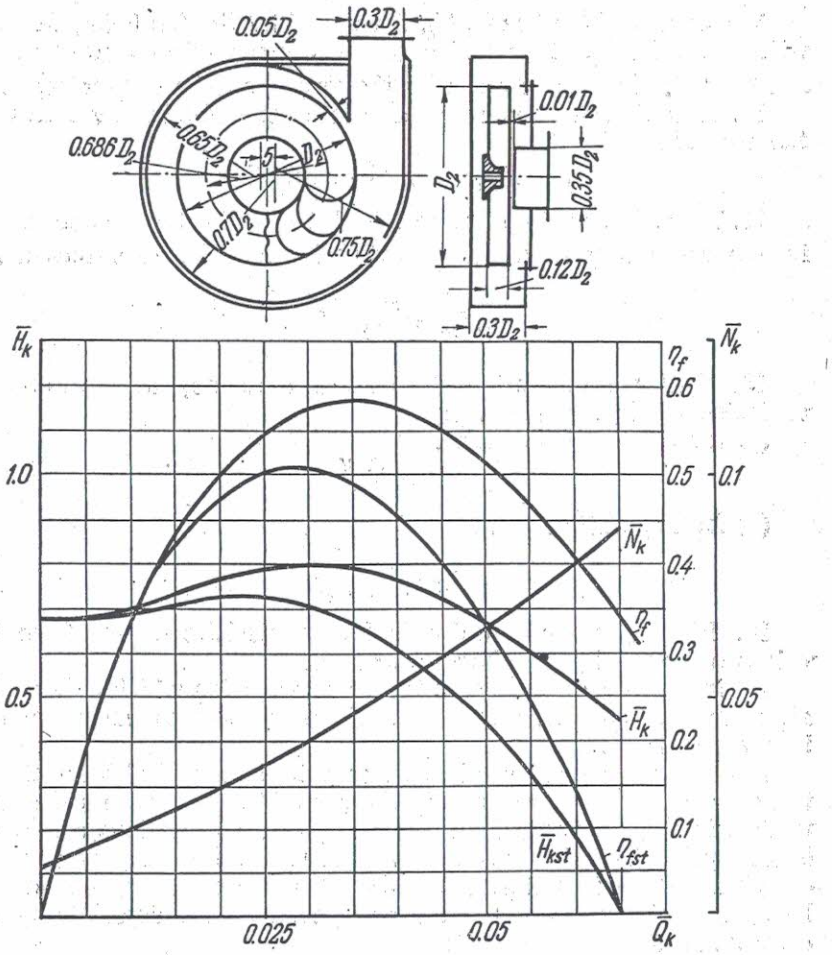
6. Using the aerodynamic diagram and the indicated dimensional proportions referred to diameter D_2 of the impeller we can compute the geometrical dimensions of the fan, setting them down on the sketch.

7. The area of the impeller is

$$F = \frac{\pi D_2^2}{4} \text{ sq m}$$

8. Next we find the discharge per second:

$$Q_{asi} = \overline{Q}_k F u_2 \text{ cu m per sec}$$



F i g. 142.

The hourly discharge, as the dimensionless characteristic, $\overline{Q}_k$, varies over the full range, will be

$$Q_{ai} = 3{,}600\overline{Q}_{ki}Fu_2 \text{ cu m per hour}$$

The obtained values of actual discharge, Q_{ai}, for the fan being designed are plotted along the horizontal axis.

9. The total head developed by the fan will be

$$H_i = \overline{H}_k u_2^2 \rho \text{ mmH}_2\text{O}$$

We compute the heads, H_i, developed by the fan being designed within the limits of variation of the dimensionless characteristic, $\overline{H}_k$, over the full range and plot the values on the vertical axis.

10. In a similar manner we calculate the static head values from the formula

$$H_{sti} = \overline{H}_{kst} u_2^2 \rho \text{ mmH}_2\text{O}$$

11. The values of the power required to drive the fan are determined in the same way from the following formula for all values of $\overline{N}_k$:

$$N_{mi} = \frac{F u_2^2 \rho \overline{N}_k}{75} \text{ hp}$$

12. Next we determine the overall efficiency in the ranges of variation of Q_{as}, H and N_a:

(a) for the total head, H_i,

$$\eta_f = \frac{Q_a H}{75 N_a}$$

(b) for the static head, H_{sti},

$$\eta_{fst} = \frac{Q_a H_{sti}}{75 N_{ai}}$$

13. Finally, we plot individual characteristic curves from the values obtained in the above procedure.

An inherent part of any fan is the volute casing which collects the air from the impeller, directs it to the outlet and converts dynamic into static head.

A discharge piece, or diffuser, may be arranged at the outlet of the volute to further recuperate the air and to reduce its velocity to that in the discharge line. According to investigations conducted by S. Rysin the diffuser should have passages diverging at an angle of 45° from the impeller. Laterally the angle of divergence should be 30°. Rysin proposes the following formula for determining the drop in head in the diffuser:

$$h_{dif} = (0.1 \text{ to } 0.15)\,\rho \frac{v_d^2}{2} \text{ mmH}_2\text{O}$$

where v_d = velocity of the stream of air entering the diffuser, m per sec.

In the Soviet Union fans are selected on the basis of diagrams which represent the results of tests conducted on all types and models of fans manufactured in the U.S.S.R. Using the head and capacity curves

of such diagrams (Fig. 143) we can find the required fan size number to ensure the highest efficiency, and the conventional speed, A. Then the overall dimensions can be found in size tables.

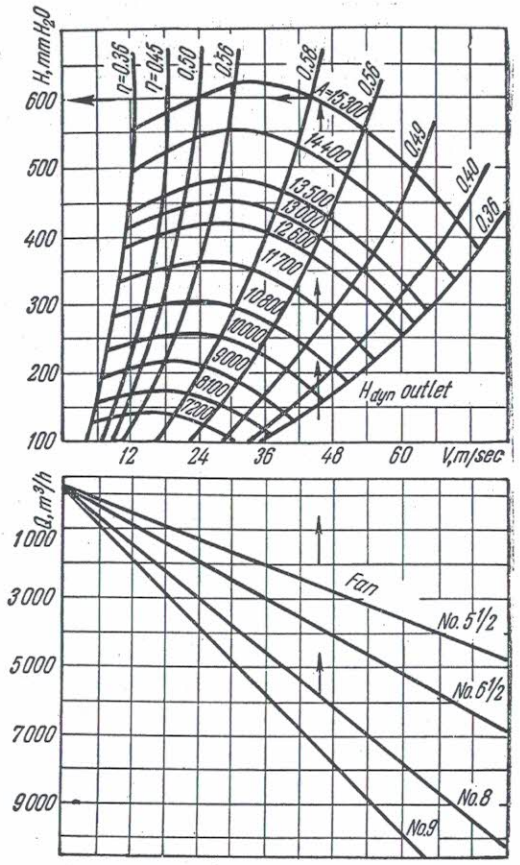


Fig. 143.

The actual speed of the fan is found from the formula

$$n_a = \frac{A}{FN}$$

where FN = fan number.

The required outlet section of the diffuser for a gas line velocity of v_{dl} is

$$F_{dif} = \frac{Q_f}{3{,}600 v_{dl}}$$

It is advisable to select the power of the motive units for fans with a certain reserve according to Table 44.

Table 44

Power on fan shaft, metric hp	Recommended power reserve, %
0.5	50
1.0	30
2.0	20
5.0	15
Over 5.0	10

Chapter 3

FLUID DRIVES

3-1. General Considerations

Fluid drives provide a flexible linkage between the motive unit and the operative member of an installation. They are designed for optimum conversion of the energy developed by the motive unit so as to ensure the highest possible efficiency in the performance of the installation.

A fluid drive is a set consisting of a centrifugal pump and a hydraulic turbine enclosed in a single casing which is filled with liquid. The pump and turbine are mounted each on the end of its own shaft. The shafts are not linked together but are in one line. The pump shaft is linked to the shaft of the motive unit, the turbine shaft to the mechanism being driven. Liquid entering the pump acquires therein the energy required to rotate the turbine wheel. Through its shaft the turbine wheel transmits this energy to the driven mechanism or the ship's propeller shaft.

Fluid drives may be classified as torque converters and fluid couplings. The former convert the torque of the motive unit and transmit the energy of this torque to the turbine shaft. Fluid couplings transmit torque from the driving to the driven shaft without conversion.

Thus, a torque converter serves both as a reducing gear and a flexible coupling linking the driving and driven shafts of the installation, while a fluid coupling serves only as a flexible clutch or coupling.

In the torque converter illustrated in Fig. 144, pump *1*, driven by the motive unit shaft *2*, draws in liquid admitted into the converter casing and discharges it into the guide vanes *3*. In the converging passages of the guide vanes static head is converted into dynamic head and, consequently, the moment of momentum of the liquid increases in reference to the axis of rotation. The liquid leaving the guide vanes enters the vanes of the turbine wheel *4* and flows towards the centre. The static and dynamic heads acquired in the pump and converted in the guide vanes, and the change in the moment of momentum of the liquid develop a torque on the shaft of turbine *4* which is transmitted through shaft *5* to the driven mechanism. The liquid

leaving the turbine wheel *4* is drawn in by the pump impeller and the process is repeated.

In a torque converter of the described type it is impossible to reverse the driven shaft if the driving shaft rotates constantly in one direction. This can be done if the torque converter is equipped with reversible guide vanes of the proper curvature or with a reversing turbine and pump.

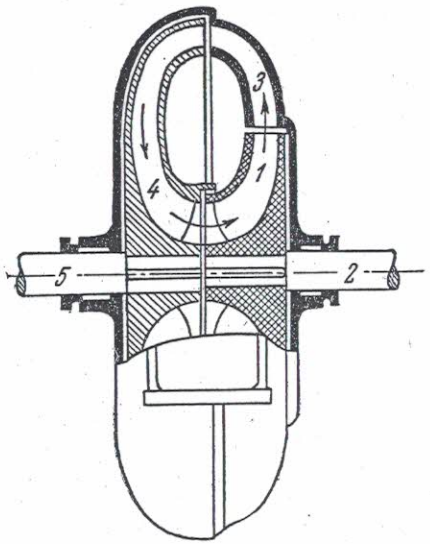


Fig. 144.

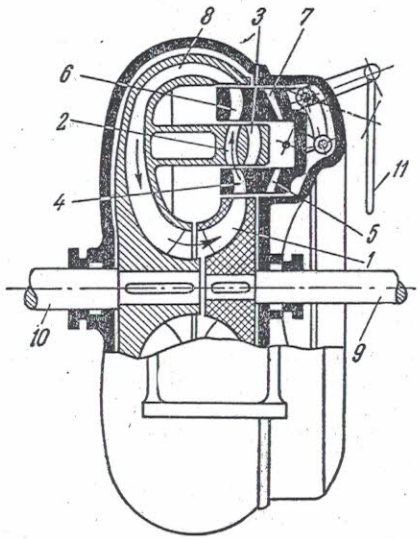


Fig. 145.

Fig. 145 illustrates a torque converter with reversible guide vanes. Pump impeller *1* in the converter is driven by shaft *9* and discharges liquid into the reversible guide vanes. After passing through guide vane passages *4* and blades *2* of the first stage turbine, guide vane passages *6* and turbine blades *8* of the second stage, the liquid again enters the pump impeller. Driven shaft *10* is rotated by the turbine wheel. To reverse the driven shaft it is necessary to change the direction of flow velocity of the liquid entering the turbine in such manner that the turbine begins to rotate in the opposite direction.

This can be achieved by shifting the guide vane unit with tie rod *11* so that passages *5* and *7* come into action.

Features of a torque converter are:

1. The driving and driven shafts are not positively engaged, being linked only by the circulating liquid in the passages of the wheel and impeller, thereby providing flexible power transmission.

This is especially important for the drives of icebreakers and ships operating in shallow waters, since the propeller may be jammed by ice or foul the bottom.

2. Torsional vibrations and impact loads are damped, creating favourable conditions for engine operation.

3. Transmission with a gearing ratio up to 12 can be accomplished between the driving and driven shafts. The speed of the driven shaft, depending upon the load acting on it, varies from that of the idling speed of the engine to zero (for instance, if the propeller is jammed by ice) with a corresponding increase in torque. At a transmission ratio from 1 to 5 the converter has an efficiency of approximately 0.9 while at a ratio from 6 to 12 the efficiency ranges approximately from 0.85 to 0.82.

4. The increase in power that can be transmitted is proportional to the product of the ratio of the speeds of the driving and driven shafts to the third power and the ratio of the pump impeller and turbine wheel diameters to the fifth power.

5. The driven shaft can be reversed with continuous uniform operation of the driving engine in the time required for one revolution (10 to 12 sec).

6. Under variable load, the driven shaft of a torque converter can readily adapt itself to the new conditions with a corresponding reduction in speed. If the driven shaft is stopped, the torque acting on it increases fourfold.

7. The shafts may be shifted up to 2 or 3 mm out of alignment due to the absence of mechanical linkage between them in the converter.

8. The torque converter has an almost constant efficiency when transmitting power to a propeller, irrespective of the load and driving shaft speed, because its characteristics closely resemble those of the propeller.

9. Torque converters can be reversed from any point on shipboard by means of a telemotor system.

10. Torque converters are distinguished for their silent operation.

11. Nonreversing engines or a group drive from several engines to a single propeller can be applied.

12. It is possible to transfer the adjustment of the running speed of the ship from the engine to the torque converter by changing the volume of liquid in the converter casing.

Torque converters enable high-speed engines to be standardized for various types of ships, the required speed being obtained on the propeller shaft by means of the converter.

The other type of fluid drive, as noted previously, is the fluid coupling (Fig. 146).

Since the fluid coupling serves only as a flexible linkage between the driving and driven shafts, its design does not incorporate a guide vane unit. The casing of the coupling comprises the driving impeller *1* and the ring *8* between whose flanges the helical ring gear *9* is secured. This gear meshes with a pinion mounted on the engine shaft. The coupling casing is fastened to its hollow trunnions *4* and *7*,

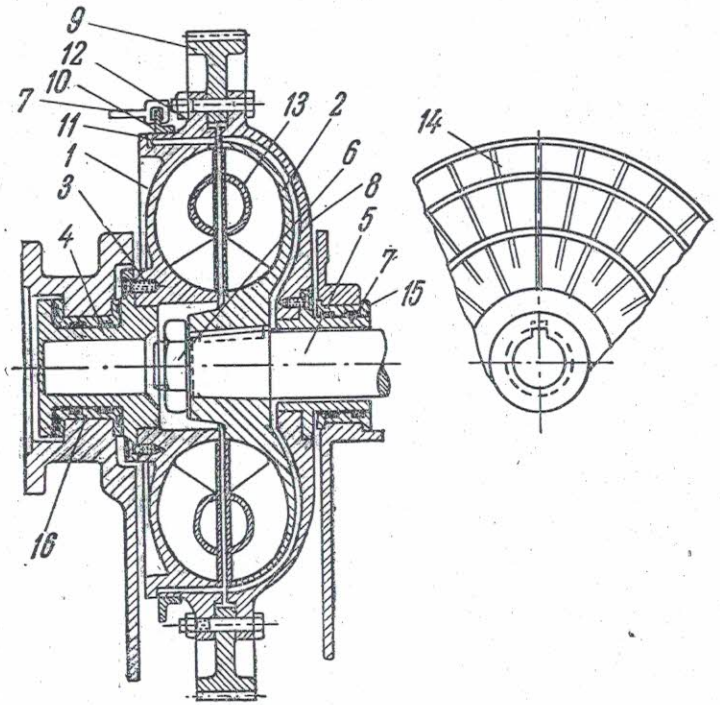


Fig. 146.

running in bearings *15* and *16*, by screws *3*. The driven turbine wheel *2* is keyed on the driven shaft *5* and is fastened by nut *6*. The back walls of the impeller and wheel, cast half-rings *13* and the radial vanes *14* form intervane passages of constant cross section. Passages *11* in the body of impeller *1* serve to drain the oil from the casing when the coupling is to be taken out of operation.

The ports of passages *11* are opened and closed by sliding ring *10* which rotates together with the impeller and is shifted along the axis of the coupling by tie rod *17*. Upon rotation of the impeller the grip *12* of tie rod *17* slides along a projecting shoulder of ring *10*. When the ports of passages *11* are closed, the coupling is filled with oil

through the hollow trunnion *4* and the gap between the pump impeller and turbine wheel. Energy is transmitted from the driving shaft in the same manner as in a torque converter. The transmitted torque, however, is not converted since there are no guide vanes. An impulse is required to force the liquid to flow from the driving impeller into the driven wheel. This impulse is developed as the result of the difference in speed of the driving and driven shafts which is due, in turn, to the slippage as the liquid passes between the impeller and wheel. The centrifugal force acting on each element of the liquid in the driven wheel is less than that in the driving impeller so that the liquid circulates from the turbine into the pump.

The slip required for coupling operation, ranging from 2 to 4 per cent, is obtained by providing a gap of 5 to 15 mm between the edges of the wheel and impeller vanes.

Either water or oil may be used as the working fluid in these couplings. Oil is preferable since it eliminates the danger of corroding the metal. Its use, however, increases the size of the coupling to some extent due to its lower density and higher viscosity. Overheating of the oil due to friction in the coupling is avoided by pumping the oil through the coupling with a special circulating pump.

Coupling action can be stopped by two methods: (1) by shifting a special shutter (ring *10*) axially to release the liquid from the coupling through holes closed by the shutter, and (2) by closing the circuit of liquid circulation with a shutter thereby rapidly disconnecting the driving and driven shafts.

Their high flexibility and dependability in operation have found fluid couplings wide application in many branches of engineering. In shipbuilding, fluid couplings are employed to connect the propeller shaft to the main engine shaft. They prove especially expedient on ships with high-speed internal combustion engines that have numerous zones of critical speeds. The elimination of a positive linkage between the engine and propeller shaft by dividing the shaft line into two independent systems excludes resonance vibrations therein to a considerable degree. This relieves the propeller shaft from overstressing due to torsional vibration.

Several high-speed diesel engines can be connected to a single propeller shaft by employing fluid couplings and gearing. The use of the couplings protects the teeth of the gearing against impacts and pulsation appearing in engine operation and, moreover, the load on the engines is automatically balanced.

A ship acquires high manoeuvrability when several engines drive one shaft through a fluid coupling and gearing, since in frequent reversals a part of the engines may be employed for forward running and the rest for reverse running, being linked to the propeller as required by engagement of the fluid couplings. Since fluid couplings

are usually reversible in their action and can transmit torque in either direction (from driving to driven side and vice versa), if one diesel is operating the rest can be started by engaging the corresponding fluid couplings.

Fluid couplings enable any engine to be cut out for repairs. When the propeller is bared in heavy seas the fluid coupling acts as a damping device preventing racing of the engine.

During towing operations in heavy and calm seas, running in shallow waters and in ice fields, a propeller driven through a fluid coupling adapts itself better to nonuniform running conditions and to jerks, preventing their transmission to the engine shaft.

Arrangements with a multiple-engine plant employing fluid couplings increase the dependability of ship operation and enable small-size engines of the same type to be used. For example, high-speed internal combustion engines with gearing and fluid couplings occupy half of the floor space and one-third of the volume taken by engines linked directly to the propeller shaft. The efficiency of a fluid coupling ranges from 0.96 to 0.98 while its approximate weight is indicated in Table 45.

Table 45

Driving shaft speed, rpm	Weight, kg/hp
1,500	0.25
400	1.5 to 2
200	6 to 8

3-2. Principal Design Equations for Fluid Drives

The main equations for fluid drive design express the relations between the heads and the power of the pump and turbine.

The theory of the centrifugal pump, taken up in a previous chapter, is applicable to the driving element (pump impeller) of a fluid drive.

(A) TORQUE CONVERTER

The theoretical head developed by the pump impeller of a torque converter is

$$H_{tp}=\frac{H_p}{\eta_p}=\frac{1}{g}(u_{2p}c_{2up}-u_{1p}c_{1up}) \text{ m} \qquad (284)$$

where H_p = actual head developed by the pump, m
η_p = hydraulic efficiency of the pump

u_{1p} and u_{2p} = peripheral velocities of the liquid at the inlet and outlet circumferences of the impeller with radii of r_{1p} and r_{2p}, respectively, m per sec

c_{1up} and c_{2up} = projections of the absolute velocities of the liquid in the direction of the peripheral velocities at the inlet and outlet circumferences of the impeller, m per sec.

The theoretical power required by the pump to develop a head of H_{tp} with a discharge per second of Q_{ts} cu m per sec through the impeller is

$$N_p=\frac{H_p\gamma Q_s}{75\eta_p} \text{ hp} \qquad (285)$$

If the angular speed of rotation of the pump impeller is ω_p the theoretical torque acting on its shaft will be

$$M_{tp}=\frac{N_{tp}}{\omega_p}=\frac{H_{tp}\gamma Q_{ts}}{\omega_p}=\frac{\gamma Q_{ts}}{g}\left(c_{2up}\frac{u_2}{\omega_p}-c_{1up}\frac{u_1}{\omega_p}\right)=$$
$$=\frac{\gamma Q_{ts}}{g}(c_{2up}r_{2p}-c_{1up}r_{1p}) \text{ kg-m} \qquad (286)$$

The theoretical head effectively utilized by the turbine wheel is

$$H_{tt}=\frac{1}{g}(u_{1t}c_{1ut}-u_{2t}c_{2ut}) \text{ m} \qquad (287)$$

where c_{1ut} and c_{2ut} = projection of the absolute velocities of the liquid in the direction of the peripheral velocities of the turbine wheel at liquid inlet and outlet

u_{1t} and u_{2t} = peripheral velocities of the liquid at the inlet and outlet circumferences of the turbine wheel with radii of r_{1t} and r_{2t}, respectively, m per sec.

The theoretical power developed on the turbine shaft with a liquid discharge of Q_{ts} per second will be

$$N_{tt}=\frac{H_{tt}Q_{ts}\gamma}{75} \text{ metric hp} \qquad (288)$$

The torque developed by the turbine when its wheel rotates with an angular speed ω_t will be

$$M_{tt}=\frac{N_{tt}}{\omega_t}=\frac{H_{tt}Q_{ts}\gamma}{\omega_t}=\frac{Q_{ts}\gamma}{\omega_t g}(c_{1ut}u_{1t}-c_{2ut}u_{2t})=$$
$$=\frac{Q_{ts}\gamma}{g}\left(c_{1ut}\frac{u_{1t}}{\omega_t}-c_{2ut}\frac{u_{2t}}{\omega_t}\right)=\frac{Q_{ts}\gamma}{g}(c_{1ut}r_{1t}-c_{2ut}r_{2t}) \text{ kg-m} \qquad (289)$$

The velocity of the liquid and, consequently, its moment of momentum $c_u r$ is either increased or decreased as it passes through

the guide vanes. It is evident from the preceding equations that, if the moment of momentum is increased, the head is also increased as is the torque acting on the turbine shaft, due to a reduction in shaft speed. If the velocity of the liquid is decreased as it passes through the guide vanes, the opposite will be true, i. e., the head will be reduced as well as the torque on the turbine shaft, since the shaft speed will increase.

The torque acting on the fixed guide vane unit will be

$$M_{gvt} = M_{1t} - M_{tp} = \frac{Q_{ts}\gamma}{g}(c_{2up}r_{2p} - c_{1ut}r_{1t}) \text{ kg-m} \quad (290)$$

The loss in head in the guide vanes is

$$h_{gv} = \left(\frac{p_{1gv}}{\gamma} + \frac{c_{1gv}^2}{2g}\right) - \left(\frac{p_{2gv}}{\gamma} + \frac{c_{2gv}^2}{2g}\right) = \frac{p_{2gv} - p_{1gv}}{\gamma} - \frac{c_{2gv}^2 - c_{1gv}^2}{2g} \text{ m} \quad (291)$$

where p_{1gv} and p_{2gv} = pressures of the liquid at inlet and outlet of the guide vanes, kg per sq m

c_{1gv} and c_{2gv} = liquid velocities at inlet and outlet of the guide vanes, m per sec.

The loss of power in the guide vanes is

$$\Delta N = \frac{Q_{ts}\gamma h_{gv}}{75} \text{ metric hp} \quad (292)$$

The efficiency of the guide vanes, referred to the actual head of the pump, is

$$\eta_{gv} = \frac{H_p - h_{gv}}{H_p} = 1 - \frac{h_{gv}}{H_p} \quad (293)$$

The work balance of a torque converter is made up of the velocity, torque and power balances.

Velocity balance. Taking all the losses in the guide vanes and in the pump into consideration by means of their efficiencies, η_{gv} and η_p, respectively, we can write the equation for the actual head developed by the pump impeller. Thus

$$H_p = \frac{\eta_{gv}\eta_p}{g}(u_{2p}c_{2up} - u_{1p}c_{1up}) \text{ m} \quad (294)$$

The same head can be expressed through the velocities of the liquid in the turbine wheel:

$$H_p = \frac{1}{\eta_t g}(u_{1t}c_{1ut} - u_{2t}c_{2ut}) \text{ m} \quad (295)$$

Combining equations (294) and (295) and making the necessary transformations we obtain

$$H_p = \frac{\eta_p\eta_{gv}}{g}(u_{2p}c_{2up} - u_{1p}c_{1up}) = \frac{1}{\eta_t g}(u_{1t}c_{1ut} - u_{2t}c_{2ut}) \quad (296)$$

and

$$\eta_p\eta_{gv}\eta_t(u_{2p}c_{2up} - u_{1p}c_{1up}) = (u_{1t}c_{1ut} - u_{2t}c_{2ut}) \quad (297)$$

Since $\eta_p\ \eta_{gv}\ \eta_t = \eta_{tc}$ is the overall efficiency of the torque converter

$$\eta_{tc}(u_{2p}c_{2up} - u_{1p}c_{1up}) = (u_{1t}c_{1ut} - u_{2t}c_{2ut}) \quad (298)$$

Torque balance. After substituting the angular speeds for the peripheral velocities in equation (298) of the summation of velocities, multiplying both sides of the equation by $\frac{Q_s\gamma}{g}$ and making all the required transformations, we obtain

$$\frac{Q_s\gamma}{g}\eta_{tc}\omega_p(c_{2up}r_{2p} - c_{1p}r_{1p}) = \omega_t(c_{1ut}r_{1t} - c_{2ut}r_{2t})\frac{Q_s\gamma}{g}$$

whence

$$\eta_{tc}\frac{\omega_p}{\omega_t}\frac{Q_s\gamma}{g}(c_{2up}r_{2p} - c_{1up}r_{1p}) = \frac{Q_s\gamma}{g}(c_{1ut}r_{1t} - c_{2ut}r_{2t})$$

and finally

$$\eta_{tc}\frac{\omega_p}{\omega_t}M_p = M_t \quad (299)$$

We shall call the product of the assigned transmission ratio, $\frac{\omega_p}{\omega_t}$, and the efficiency of the converter, η_{tc}, the coefficient of torque conversion, $k_{tc} = \eta_{tc}\frac{\omega_p}{\omega_t}$. Since for actual torques

$$M_t = k_{tc}M_p \neq M_p \quad (300)$$

the change in energy of the liquid, needed to convert the torque of the pump impeller, occurs in the guide vanes. The additional torque developed in the guide vanes will evidently be

$$M_{gv} = k_{tc}M_p - M_p = M_p(k_{tc} - 1) = M_p\left(\eta_{tc}\frac{\omega_p}{\omega_t} - 1\right) \quad (301)$$

Therefore, the total moment transmitted to the turbine is

$$M_p + M_{gv} = k_{tc}M_p = M_t \quad (302)$$

i. e., the torque of the turbine wheel is equal to the sum of the torques of the pump impeller and guide vanes.

Power balance. The power, N_p, required by the pump, taking into account the losses in the guide vanes, and the power, N_t, developed by the turbine are

$$N_p = \frac{Q_s \gamma H_p}{75 \eta_p \eta_{gv}} \quad \text{and} \quad N_t = \frac{Q_s \gamma H_p \eta_t}{75} \tag{303}$$

It follows that

$$H_p = \frac{75 N_p \eta_p \eta_{gv}}{Q_s \gamma} = \frac{75 N_t}{Q_s \gamma \eta_t} \tag{304}$$

The power developed by the turbine wheel and that in the pump are equal. Thus

$$N_p \eta_p \eta_{gv} \eta_t = N_t = N_p \eta_{tc}$$

$$\eta_{tc} = \frac{N_t}{N_p} \approx 0.82 \text{ to } 0.90 \tag{305}$$

The results obtained from these work balance equations of a torque converter serve as the initial data in the design of fluid drives.

(B) FLUID COUPLINGS

By analogy with the torque converter we can write the work balance of a fluid coupling in the following way:

Velocities:

$$\eta_p \eta_t (u_{2p} c_{2up} - u_{1p} c_{1up}) = (u_{1t} c_{1ut} - u_{2t} c_{2ut}) \tag{306}$$

Torques:

$$\eta_p \eta_t \frac{\omega_p}{\omega_t} M_p = M_t \tag{307}$$

Since there is no conversion of energy in guide vanes in fluid couplings, $M_p = M_t = M_{sh}$, and thus we can write that

$$\eta_p \eta_t = \frac{M_t \omega_t}{M_p \omega_p} = \frac{\omega_t}{\omega_p} = \frac{n_t}{n_p} = \eta_{fc} \tag{308}$$

where η_{fc} = efficiency of the fluid coupling.

Powers:

$$N_p \eta_t \eta_p = N_t = N_p \eta_{fc} \text{ or}$$

$$\eta_{fc} = \frac{N_t}{N_p} \tag{309}$$

Therefore, the efficiency of the fluid coupling is

$$\eta_{fc} = \frac{\omega_t}{\omega_p} = \frac{n_t}{n_p} = \frac{N_t}{N_p} \tag{310}$$

If the lag in the turbine shaft speed, n_t, behind the pump shaft speed, n_p, referred to the latter, is called the slip, S, the efficiency, η_{fc}, can be additionally expressed as follows:

$$\eta_{fc} = \frac{n_t}{n_p} = \frac{(n_p - n_p + n_t)}{n_p} = 1 - \frac{n_p - n_t}{n_p} = 1 - S \tag{311}$$

In the same way as for a torque converter, the preceding balance equations serve as the basis for fluid coupling design.

Part Two

SPECIALIZED AUXILIARY MACHINERY

Chapter 4

STEERING GEAR MECHANISMS

4-1. Purpose and Components of Steering Gears

The steering gear, whose purpose is to ensure ship control and manoeuvring quality, consists of:

1. The rudder to which forces are applied to change the direction of the ship's motion.
2. The steering gear which links the rudder to the steering engine.
3. The steering engine, comprising the gearing and engine proper which power steering gear operation: putting over and locking the rudder, and indicating its position.
4. The control gear (teledynamic transmission) of the steering engine which links the latter with the control post of the ship.

Rudders may be positive-action or passive in their mode of operation; they may be simple, semibalanced or balanced in design. Passive operation rudders may be either of the flat or hydrofoil type.

Positive-action rudders are all of the hydrofoil type. Rudders are put over from side to side by means of:

(1) tiller gear with a steering rope;
(2) quadrant drive with toothed gearing;
(3) drives with the steering engine mounted on the rudder;
(4) screw-type steering gear;
(5) hydraulic gear.

These drives are powered by:

(1) hand steering gear;
(2) steam-driven steering engines;
(3) electrical steering gear;

(4) hydraulic gear (pumps).

Steering engines are started, stopped, reversed and their shaft speeds are changed from the control station by means of a teledynamic transmission which links the steering wheel with the starting mechanisms of the steering engine. In this manner the speed and direction of rudder motion are controlled.

Teledynamic transmissions are classified as:

(1) Mechanical:
(a) shaft type, (b) steering rope and (c) rack type.
(2) Hydraulic.
(3) Electrical.

The following requirements characterize the features that must be provided by all steering gears:

(1) Dependable and unfailing operation under any navigating conditions.
(2) Long service life.
(3) Ability to put the rudder over to the required angle at full sea speed.
(4) Ability to put over the rudder at the required speed of rudder motion.
(5) Possibility of rapid change-over from the main type of steering to the auxiliary facilities.
(6) Possibility of control from several places on shipboard.
(7) Convenience in steering.
(8) Minimum overall size and weight.
(9) Simple design, maintenance and servicing.
(10) Economical operation.

Steering engines and control gear must satisfy the additional requirements given below.

(1) The steering engine should be started by a steering wheel in the main and standby control stations.
(2) The steering engines should be reversible so that they can put the rudder over in either direction.
(3) The direction of rotation of the steering engine should be such that the rudder is put over in the same direction that the steering wheel is turned. When the wheel is stopped, the steering machine must stop the rudder whose position is shown by a rudder indicator.
(4) The steering engine must actuate the control gear and the rudder immediately in the required direction as soon as the helmsman begins to turn the wheel, so as to ensure uninterrupted control of the ship.
(5) The steering engine must stop automatically when the rudder reaches its hard-over positions to prevent damage to the steering gear.

(6) The speed with which the rudder is put over must correspond to that with which the wheel is turned so that the desired rapidity of manoeuvring can be achieved.

(7) The steering engine must operate silently and must be properly balanced so that its operation does not lead to hull vibration and is in no way detrimental to the living conditions of the crew and passengers.

These requirements are met by proper design of the steering engine and control gear.

According to the regulations of the U.S.S.R. Shipping Register, the main set of steering gear of ships 60 m and more in length must be actuated by a mechanical motive unit. Hand steering gear may be installed on shorter ships if the rudder can be put from hard-over to hard-over in 30 sec at full sea speed by one man exerting not over 12 kg on the wheel rungs and turning the rungs at a peripheral speed not exceeding 0.8 m per sec. In passenger ships with a rudder-head shaft over 225 mm in diameter, the standby control gear should be powered from a mechanical motive unit.

Hand standby steering gear may be employed in all other cases if the effort to be exerted by one helmsman on a rung does not exceed 16 kg at a peripheral speed of the rungs not over 1 m per sec and if not more than four steersmen are required.

The time required to put the rudder over from 20° on one side to 20° on the other with the standby gear must not exceed 60 sec at one half of the full sea speed, but not less than 6 knots. The time required to put the rudder from hard-over to hard-over is not stipulated.

The design of the steering gear must provide for a change-over from the main to the standby arrangements in two minutes or less.

The steering gear should incorporate a properly coordinated system limiting the amount that the rudder can be put over.

The stops for the tiller or quadrant should be installed on the deck in such manner that they allow the rudder to be put over 1.5° beyond the stops on the steering engine.

Additional stops may be provided on the rudder-post allowing a rudder angle of 1.5° more than that allowed by the deck stops.

The stops installed on the steering engine should correspond to a rudder angle of 35° for flat rudders and 32° for the hydrofoil type. Provision is made for a braking device to hold the rudder in any given position, if necessary. This device is designed to handle a torque one-fourth of the maximum value on the rudder-head shaft. The braking device must be capable of holding the rudder at any possible angle to the middle-line plane at one-half of the normal speed of the ship, but not less than 6 knots.

In the great majority of transport ships the steering engines are located:

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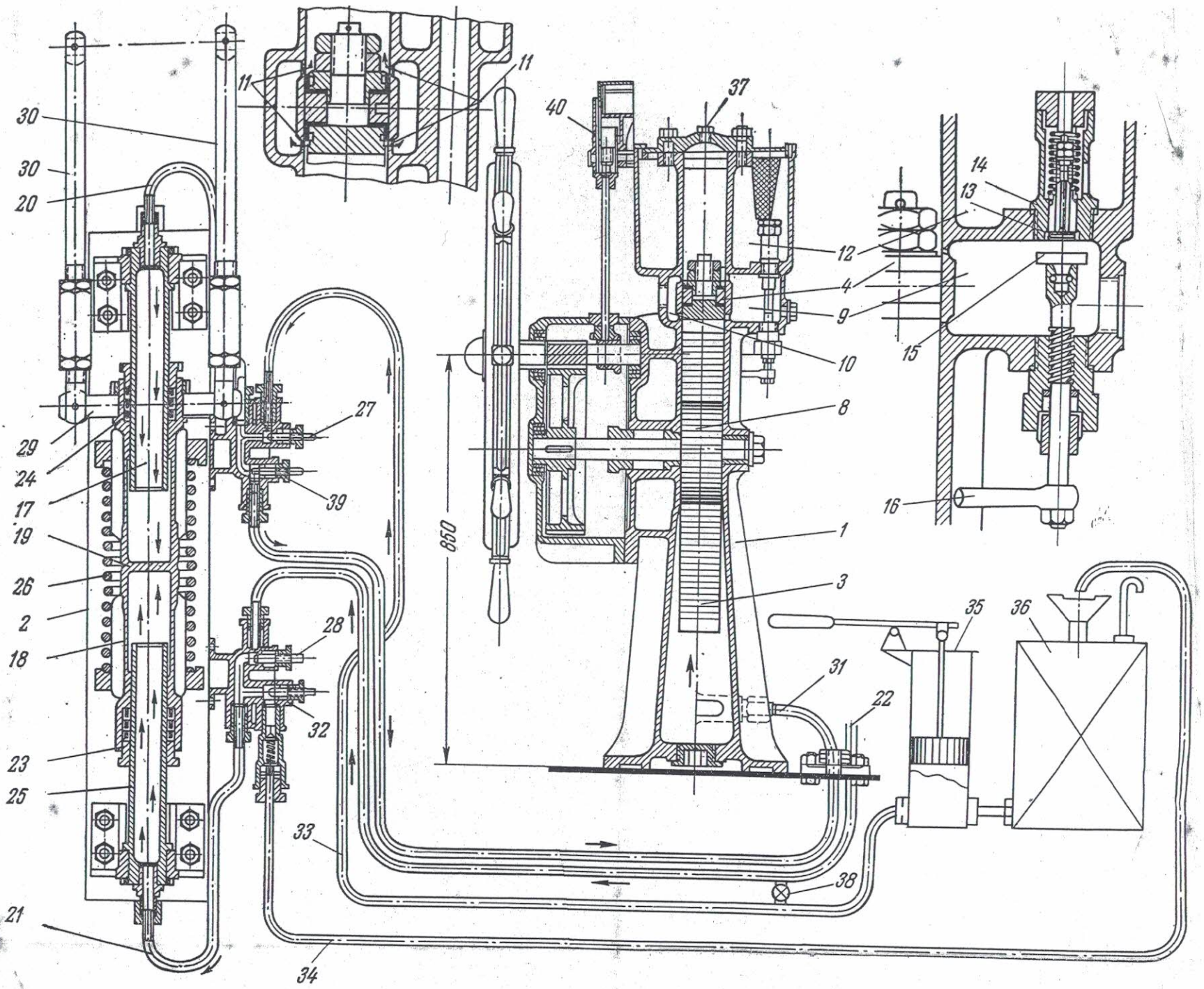

Fig. 147

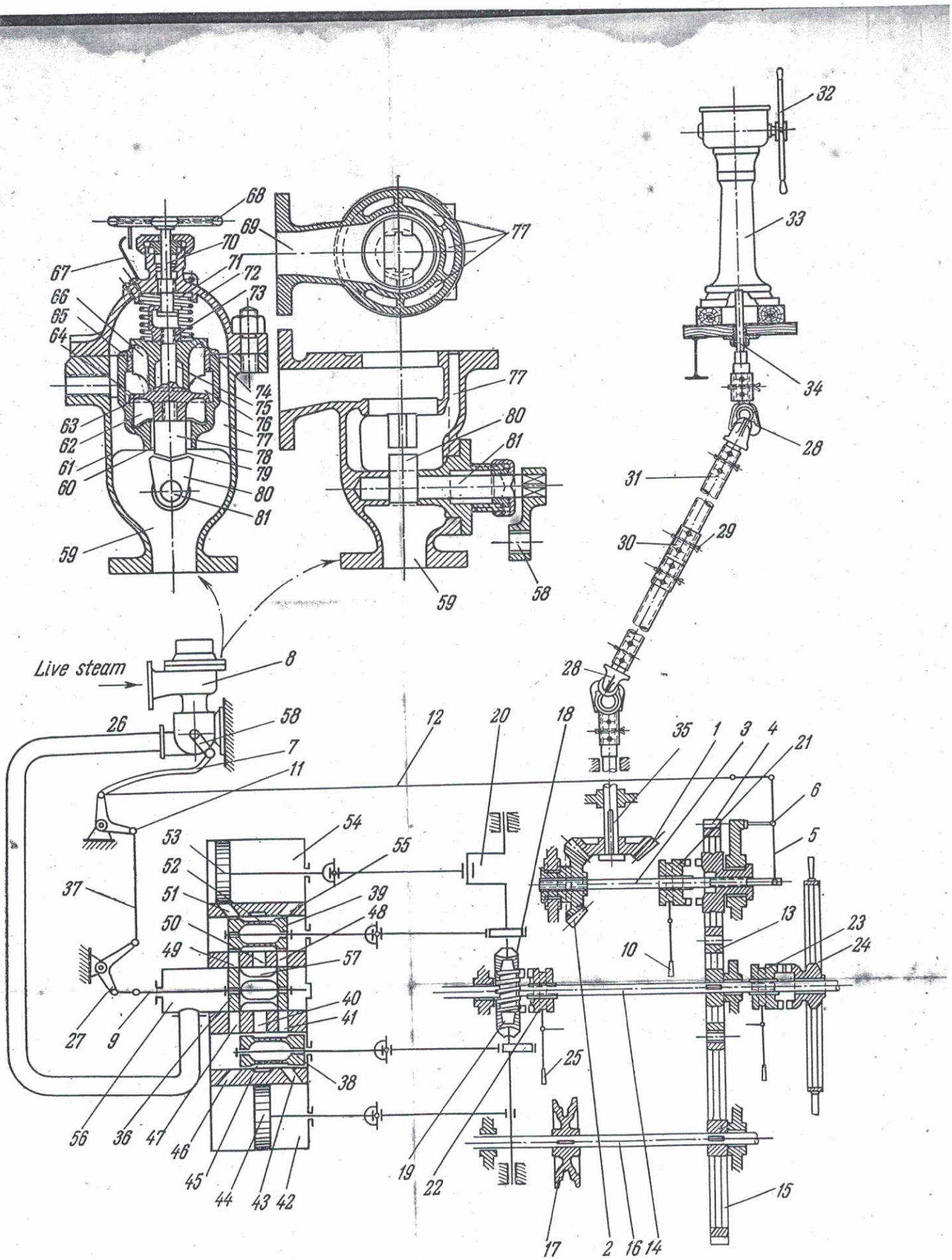
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Fig. 148

(1) at the stern bulkhead of the engine room at the level of the main deck;

(2) in the tiller room in the ship's stern directly adjacent to the rudder-head shaft;

(3) near the main control station of the ship, either in the wheelhouse or directly under it.

Let us appraise these three possible locations of the steering engines on amidships-engined ships from the point of view of:

(1) convenience in servicing the steering engines;
(2) ensuring their dependable operation;
(3) ensuring reliable action of the steering gear;
(4) ensuring reliable operation of the teledynamic transmission;
(5) steam consumption;
(6) facilities for convenient and rapid repairs at sea or in harbour;
(7) their effect on the living conditions of the crew and passengers.

It may be noted that the first of the three possible locations of the steering engines has substantial advantages over the second and third as to servicing convenience, dependability of operation, accessibility for repairs at sea and in harbour, steam consumption, and in the insignificant disturbance occasioned by the noise of the engines since they are far from the living quarters.

As to the reliability of the linkage between the steering engine and the rudder-head shaft, the first location is inferior to the second because of its long kinematic chain between the engine and the tiller leading to a reduction in the efficiency of the steering gear as a whole.

Location of the steering engines in the tiller room is a more dependable arrangement of the linkage with the rudder than in the first case. The steam consumption in the steering engine will be somewhat higher in this case due to more condensation in the longer steam line. This will be especially significant for navigation in northern latitudes. Moreover, the transmission from the wheel to the engine will be longer, leading to less reliability and accuracy in the action of the whole steering gear. Servicing convenience is achieved with the steering engine arranged adjacent to the navigation bridge, as well as more dependable linkage with the steering wheel and more rapid reaction to movements of the wheel.

Steering engines with rope transmission are installed next to the stern bulkhead of the engine room or near the main control station; those with quadrant, screw-type or hydraulic drives are located in the tiller room at the ship's stern.

Steam-driven steering engines were previously employed on ships with a steam power plant, while the electrical and electro-hydraulic gears were used on motorships. Recently, however, the development of heat utilization of the waste gases of internal combustion engines

in exhaust boilers, fired in harbour with liquid fuel, and the installation of self-contained auxiliary boilers have made feasible the equipment of motorships with steam-driven deck machinery, including steam-driven steering engines.

At the same time, steamships are frequently equipped with electrical and electro-hydraulic steering gear powered from economical steam- and turbo-dynamo installations. Steam-driven steering engines are powered by either horizontal or vertical steam engines.

The advantages of electrical and hydraulic steering engines in comparison with the steam type are:

(1) convenience in installation and more reliable linkage to the control station (electric wiring instead of telemotors);

(2) high sensitivity of the steering controls;

(3) readiness for action at any time without any preparatory operations.

Disadvantages of electrical steering gears include their higher cost, and more complicated adjustment and operation. Steam-driven steering engines, in their turn, possess the following advantageous features when compared to the electrical type:

(1) more dependability;

(2) simple construction;

(3) lower initial cost;

(4) less transmissions.

In comparison with electrical steering engines, the steam type:

(1) are less economical, having a higher steam consumption;

(2) have less sensitivity in steering control;

(3) have greater overall size;

(4) require timely preparation (heating) of the engine before it can be put into operation;

(5) require more complicated maintenance in operation.

The favourable features of a hydraulic steering engine include the high sensitivity to movement of the steering wheel or control lever, its smooth, silent operation, constant torque developed on the rudder-head shaft, wide range of adjustment of rudder motion speeds, safety in operation and a higher efficiency than with other types of engines.

The overall size and weight of hydraulic steering engines are less than those of the electrical and steam types, even though they develop a higher torque on the rudder-head shaft.

Disadvantages of hydraulic steering engines are their more complicated manufacture, installation and adjustment, high initial cost, and the fact that they require especially careful maintenance and servicing in operation.

4-2. Principles of Steam Steering Engines

Reversibility and the capability of starting the steering engine shaft from any position are achieved by employing two-cylinder engines without steam expansion and with the crankpins of the crankshaft arranged at 90°. According to steam engine theory, if the valves have no lap and the eccentricity is at 90° to the crankpin, reversing can be accomplished by changing steam admission into the cylinder from external to internal and vice versa.

The valve gear and reversing of all steam steering engines are based on this principle.

The shaft of a steam steering engine is started, stopped and its rotational speed is varied by means of mechanical, hydraulic or electrical teledynamic transmissions.

(A) TELEDYNAMIC TRANSMISSIONS *

The following types of mechanical teledynamic transmissions are employed.

Rod-type transmissions usually consist of a series of steel rods and hollow shafts from 40 to 60 mm in diameter. The rods are linked together with couplings, bevel gearing and universal joints to enable the gear to be adapted to the local conditions on the ship. As a rule, the rods pass under the deck. Mounted in underhung bearings between the steering pillar and the starting device of the steering engine, the rod transmission is actuated by the steering wheel and transmits its motion to the starting valve. The distance between the rod supports should not exceed 70 diameters for solid rods or 80 diameters for hollow shafts.

Such controls may be used when the distance between the steering engine and the wheel is relatively short. It is inadvisable to use rod-type transmissions over considerable distances due to the danger of jamming in the system resulting from deformation of the deck or bulkheads. Rudder motion will not exactly coincide with the turning of the wheel because of the inevitable backlash in the bevel gears. Considerable force must be exerted on the wheel to rotate the rods.

Bar and rack transmissions consist of bars resting on rollers and moving axially in either direction when the wheel is turned to actuate the starting valve. The first bar is linked to the wheel through a rack meshing with a pinion driven by the wheel. The force exerted on the first bar by the rack is transmitted to the following bars. The last bar is linked to the starting valve by a second rack and pinion

* Electrical telemotors are taken up in the study course "Marine Electrical Equipment".

arrangement. This type of transmission is usually installed on self-propelled barges and on ships having ample room on the upper deck to accommodate it.

A drum-type transmission may be employed both for short and long distances between the wheel and steering engine. It consists of two drums, one linked to the wheel and the other to the starting valve. The drums are linked together with a flexible wire rope from 8 to 12 mm in diameter. An advantage of this system is the ease with which it can be installed, while its chief disadvantage is sag in the rope (leading to excess play in the transmission) due to its stretching. Therefore, provision must be made for taking up the sag if this type of transmission is to be used.

A hydraulic transmission (telemotor control) consists of the transmitter which is a cylinder and piston actuated by the steering wheel and a receiving motor which also is a cylinder and piston that actuate the starting valve through a lever system. The two cylinders are connected together by small-diameter piping.

The piping and cylinders are filled with oil or a mixture of glycerine and water with a low freezing point. This liquid transmits the motion of the wheel and the transmitter piston connected to it to the starting valve.

One construction of a hydraulic telemotor controls, consisting of transmitter *1* and receiving motor *2*, is shown in Fig. 147. The transmitter casing houses the piston *4* with gear rack *3* meshing with pinion *8*. The latter is driven from the wheel through gearing.

In the central position of the piston the upper and lower ends of the transmitter cylinder are connected by passages *11* with the spaces *10* and *9* outside of the jacket at the middle of the cylinder.

Space *9* is connected through valves *13* and *14* with the replenishing tank *12*. Valve *13* serves to replenish the system with liquid while relief valve *14* releases excess liquid from the system. When the system is charged with liquid, replenishing tank *12* is separated from space *9* by disk-valve *15* which shuts the ports of valves *13* and *14* when handle *16* is turned.

The receiver *2* consists of the hollow rams *17* and *25* fixed on the bulkhead and entering the upper and lower parts of cylinder *18*, which is divided by partition *19* into two independent cylinders. The lower end of the transmitter cylinder is connected by pipes *31* and *20* to the upper ram *17* and, consequently, to the upper part of cylinder *18*. Pipes *21* and *22* connect the upper end of the transmitter to the lower ram and, through it, to the lower part of the receiver cylinder.

This arrangement of the receiver enables leaks to be eliminated in the moving stuffing boxes *23* and *24* of the rams since the boxes are accessible for inspection and repairs. Spring *26* brings cylinder *18* to its midship position when the steering wheel is released. Charging valve *27* serves to connect the system with pipe *33* which runs from the hand charging pump *35* and is opened when the system is filled from the charging tank *36*. Valve *32* serves to drain liquid from the system along pipe *34* into the charging tank *36*, from which it is drawn by hand pump *35* in charging the system.

Circuit valves *39* and *28* disconnect the receiver from the telemotor system.

Crosshead *29* and connecting rods *30* transmit the motion of the receiver cylinder body *18* to the starting device of the steering engine.

Let us assume that piston *4* is raised by the action of the steering wheel, thereby discharging liquid through pipe *22* into the lower ram *25* of the receiver, and forcing cylinder body *18* upwards. From the upper part of cylinder *18* liquid will be displaced through pipe *31* into the lower end of the transmitter cylinder. In its upward movement cylinder body *18* compresses spring *26* and moves crosshead *29* and connecting rods *30* in the same direction to actuate the starting device of the steering engine. If the helmsman stops turning the wheel but does not release it, all the moving parts of the arrangement also stop. The servomotor also stops the steering engine after the rudder has been put over to a certain angle. If the helmsman continues to turn the wheel in the same direction the rudder continues to move towards the same side. When the helmsman releases the wheel, cylinder body *18* of the receiver is returned to the midship position by the action of spring *26*, thus displacing liquid from ram *25* along pipe *22* into the upper end of the transmitter cylinder, and returning piston *4* to its midship position. The hand *40* of the rudder indicator is also actuated through gearing. At the same time, the steering engine is started and it puts the rudder over into the middle-line plane of the ship. As soon as the piston reaches its central position and the rudder is in the middle-line plane of the ship, the steering wheel stops rotating and the steering engine stops operating.

If the telemotor controls are properly adjusted, the indicator hand *40* shows 0° and piston *4* and cylinder body *18* are in their midship positions. Any other positions of the steering gear components indicate that the telemotor controls are out of order and must be repaired or adjusted without delay. Valve *38* is used to connect the upper and lower parts of the receiver when the telemotor is out of order.

The telemotor system is charged with liquid with piston *4* and cylinder body *18* in their midship positions, with plug *37* removed and valves *27*, *28*, *32* and *39* opened.

As the system is being charged by hand pump *35*, air will be purged through the hole that is usually closed by plug *37*. As soon as liquid begins to run out of this hole, plug *37* is replaced and pumping

is continued until liquid begins to flow from drain pipe *34*. Noting the time *t* that expires from the beginning of charging until liquid starts to run from the drain pipe, pumping is continued for a time equal to approximately 2*t*. The working pressure in the telemotor system varies from 2 to 14 kg per sq cm.

The telemotor system is charged either with high-quality mineral oil having a viscosity of 3 or 4 Engler degrees at 50°C or a mixture of glycerine and water.

The data of Table 46 may be recommended in making up a non-freezing mixture of glycerine and water.

Table 46

Glycerine content of the mixture, %	Fit for use at temperatures down to, °C
10	—6
20	—12
25	—15
30	—17
35	—25
50	—30

The advantageous features of hydraulic telemotor controls are:

(1) precise transmission of wheel motion to the starting valve;

(2) unfailable operation;

(3) no influence of deformation of the ship's hull on the operation of the telemotor controls;

(4) convenience in installation;

(5) automatic return of the rudder to the middle-line position.

Some of the disadvantages are:

(1) careful maintenance and servicing are required;

(2) the equipment is heavier than that of other transmissions;

(3) more complicated adjustment is required than for other types.

(B) VALVE GEAR, SERVOMOTORS AND ECONOMIC VALVES

Next we shall consider the valve gear and interaction of the parts of a steering engine in operation as shown schematically in Fig. 148. The steering gear is actuated by the steam engine when clutch member *22* is engaged by shifting lever *25*.

Rotation of the steering wheel *32* is transmitted through gearing mounted in pillar *33*, shaft *34* and the rod-type transmission to the final shaft *34*. The transmission consists of universal joints *28*, cotter pins *29*, hollow shafts *31* and couplings *30*. Rotation of shaft *35* is

transmitted through bevel gears *1* and *2* to shaft *3*, on which gear *2* is mounted on a feather. Shaft *3* rotates and moves axially at the same time since its right end is threaded and screwed into the hub of gear *4* which is restricted in the axial direction. This axial movement of shaft *3* turns lever *5*, pivoted to the shaft, about pin support *6*. Lever *5* actuates tie rod *12* and this turns angle lever *11*. The latter, in turn, actuates tie rod *37* and turns angle lever *27* to shift the starting valve *36* from its central position through valve rod *9*.

Let us assume that the starting valve is shifted to the right. Then live steam, admitted through the economic valve *8* and pipeline *26* into valve chest *56* of starting valve *36*, enters passages *47* and *49*. This constitutes the external admission of steam. Steam-distribution valve *39* closes the port of passage *49* so that no steam is admitted from there. Steam from passage *47* passes through the chest of valve *38* and enters the left end of cylinder *42*, forcing piston *44* to the right to rotate the crankshaft *20* of the steam engine in the clockwise direction. Spent steam is exhausted from the right end of cylinder *42* along passage *43* into cavity *45* and from there through passage *40* into cavity *57* which is connected to the exhaust steam line. By means of the eccentric sheave linkage, the rotating crankshaft shifts steam-distribution valve *39* from its central position and the steam engine begins to run. Worm *19*, meshing with worm wheel *18*, rotates together with the crankshaft. The worm wheel is mounted freely on shaft *14* but drives it when clutch member *22* is engaged. Rotation is transmitted further through gears *13* and *4*, gear *15* and shaft *16* to sprocket *17*. The latter actuates the wire rope connected to the tiller which, in turning, puts over the rudder. Rotation of gear *4* shifts shaft *3* axially in the direction opposite to that in which it was moved when the steering wheel was turned.

If the steering wheel is turned at a uniform rate of speed, the starting valve *36* remains stationary and the engine also runs at a uniform rate. This is due to the fact that the speeds of axial movement transmitted to shaft *3* in one direction by gear *2*, and in the opposite direction by gear *4* are equal to each other.

When the helmsman begins to turn the wheel faster, the speed of axial motion transmitted to shaft *3* by gear *2* will be greater than that transmitted by gear *4* in the opposite direction. In this case, starting valve *36* opens the ports of the chest to a greater degree and the engine crankshaft begins to rotate faster. Consequently, the rudder also turns faster.

If the helmsman turns the wheel slower, then the speed of axial movement transmitted to shaft *3* by gear *2* will be less than that transmitted by gear *4*. As a result, the starting valve *36* closes the ports of its chest to a greater degree, the crankshaft begins to rotate at a lower speed and the rudder is put over slower.

Let us suppose that the helmsman ceases to turn the wheel. In this case, the rotation of gear *4* returns shaft *3* to its initial position where it shuts the economic valve and returns the starting valve to its central position by means of the levers and tie rods described above.

As the starting valve approaches its central position the effective section of the steam admission ports is reduced, thereby decreasing the amount and pressure of the steam entering the cylinders. Finally, when the pressure of the steam in the cylinders has been reduced to a value so low that the force exerted against the pistons of the steering engine can no longer overcome the resistances of the steering gear, the engine stops. Consequently, the elements comprising the gear train of the servomotor stop as well, as does the starting valve, before it reaches its central position. If the engine is not equipped with an economic valve or if it is not closed when the engine stops, live steam will continue to enter the cylinders through the slots left in the ports and begin to condense there.

If the engine is now started in the opposite direction by turning the steering wheel, this steam in the cylinders will be let out into the exhaust line without doing any useful work. Since the crankpins of the engine are arranged at 90°, the engine may contain approximately one cylinder volume of live steam when it is stopped. If we take into consideration the fact that the steering engine is reversed, on an average, up to 10 times per minute, we see that 600 cylinder volumes of steam are wasted per hour. This, in conjunction with the absence of steam expansion in the cylinders, are the reasons why steering engines have such a high steam consumption.

This wasteful steam consumption is made the worse by the fact that the steering engine becomes less sensitive to the action of the starting valve when it begins to operate. This is due to the mixing of the live steam with the steam remaining in the cylinder, which leads to a reduction of the mean indicated pressure in the cylinder at the initial moment when the engine is started.

Steering engines are equipped with an economic valve *8* which closes when the engine is stopped and cuts off steam admission to the chest of the starting (differential) valve.

Let us suppose that the starting valve is shifted to the left when the steering wheel is turned. In this sase, live steam passes through the hollow starting valve *36*, passages *41* and *48*, cavity *45* of steam-distribution valve *38* and passage *43* into the right end of cylinder *42*. Internal steam admission is accomplished in this manner. The pressure of the steam forces piston *44* to the left, rotates the crankshaft in the counterclockwise direction by means of the crankpin and shifts valve *39* from its central position by means of the valve gear.

Spent steam is exhausted from the left end of cylinder *42* through passage *46*, the left end of the valve chest of valve *38*, passage *47* and cavity *57* which is connected to the exhaust steam line.

Steering engines have a device for putting over the rudder by hand. This is done with the aid of the telemotor and steering wheel *24*.

To steer by hand through the telemotor, clutch member *21* is shifted by lever *10* into engagement with gear *4* to exclude axial motion of shaft *3*. Besides this, clutch member *22* and worm wheel *18* are disengaged. Now sprocket *17* is rotated by shaft *3* through gears *4*, *13* and *15*

The rudder can be put over by hand from wheel *24* only after additionally engaging clutch member *23* with wheel *24* and disengaging shaft *3* from bevel gear *2* by removing the feather.

Spent steam is exhausted from cylinder *54* through passages *50*, *51* and *55* and cavity *52*.

It follows from the above that:

(1) coordination of steering wheel and steering engine operation is accomplished by a system of transmissions linking the wheel, starting valve and engine shaft, this system being called the servomotor;

(2) the operation of the servomotor is based on the principle of adding the length of travel of its components during steering engine operation;

(3) a steering engine must be equipped with an economic valve to reduce steam consumption; this valve must: (a) automatically cut off live steam admission to the cylinders when the engine comes to rest, and (b) ensure tightness when closed (from the moment the valve disk reaches its seat);

(4) the valve should close on its seat after the starting valve is stopped, independently of the steering engine, by the mechanism of the valve itself;

(5) the design of the economic valve should be as simple as possible.

Economic valves are classified as mechanically and steam operated types. Mechanical economic valves are actuated by a mechanical device; steam valves by the pressure of the steam.* The system illustrated in Fig. 148 is equipped with a mechanical economic valve. Body *61* houses the double-disk valve. Shank *78* with a tapered lower end *79* is screwed into the lower disk *64*. The upper disk *74* is fitted freely on the stem *75* of the lower disk *64* and is held against the lower disk by bushing *73* screwed on the end of the valve stem. Both valve disks have guide wings *62* and *66*. Spring *71*, provided between the cover *72* of the valve body and the upper

* Steam operated economic valves will be taken up together with the steam steering engine for drum-type transmission (see Figs. 151 and 152).

disk *74*, holds both disks on their seats *65* and *63*. A shoulder at the end of stem *70* of handwheel *68* is fitted into bushing *73*. Both valve disks can be raised from their seats by turning the handwheel. This is done to cut out the economic valve, thereby bypassing it in admitting steam to the starting valve. The lift of the valve disks above their seats can be checked by means of indicator *67*. Shank *78* is arranged above lug *80* which is of dovetail shape. A clearance of 0.5 to 2 mm is provided between the tapered end of the shank and the dovetail of the lug to allow the disks to drop to their seats at the proper time.

Lug *80* is mounted on shaft *81*, as is single-arm lever *58* which is actuated in the same way as the starting valve by the telemotor through lever 7. Live steam enters the economic valve through connection *69*, being admitted into space *76* between the disks *74* and *64*. When the disks are lifted steam passes through passages *77* and *60*, connection *59* and the piping into the starting valve. The valve disks drop to their seats under the action of spring *71* before the starting valve reaches its central position. Two-seat valves are employed because of their balanced operation, though their disks are more difficult to fit to the seats than in the single-seat type.

As previously mentioned, a servomotor links the steering wheel and starting valve through transmissions with the steering engine shaft, and its purposes are to stop the steering engine at the required rudder angle and to coordinate the speed of wheel rotation with that of rudder motion.

The steering engine shown schematically in Fig. 149 is equipped with another type of servomotor. Here, the steam engine with the cylinders *1* (the second cylinder is behind the plane of the drawing) drives the crankshaft of the steering engine, made integral with worm *4*, through piston rod *2* and connecting rod *3*. Rod *2* and connecting rod *3* are linked to a crosshead having a shoe *5* that slides along guide *6* during engine operation.

Worm *4* meshes with worm wheel 7, mounted on shaft *8*. The shaft is fixed against axial movement by the shoulders *10* of the journal running in bearing *9*. The left end of shaft *8* has a thread *11* on which the threaded hub *12* of steering wheel *13* is mounted. Hub *12* has an annular groove into which the trunnions *14* of the forked angle lever *15* enter without impeding rotation of the hub. Angle lever *15* is pivoted on pin *16*. Tie rod *17* of the starting valve in valve chest *18* is linked to the other end of the angle lever. Tie rod *17* is linked to stem *19* of the starting valve. Drum *20*, on which several turns of the tiller rope are wound, is mounted on shaft *8*.

This steering engine operates as follows. When the helmsman starts to turn the wheel, hub *12* will either be screwed on or off the threaded part *11* of shaft *8*, since the shaft of the engine, worm *4*,

worm wheel 7 and shaft *8* are at rest. This axial motion of the hub along the shaft turns lever *15* which shifts the starting valve from its central position by means of the tie rod to start the engine. The crankshaft and worm begin to rotate in the same direction as the steering wheel. Worm *4* rotates worm wheel 7 and, consequently, shaft *8*. The rope drum also begins to rotate and the tiller rope *21* actuates the tiller which puts over the rudder.

The threaded hub *12* of steering wheel *13* will begin to move axially along the shaft in the direction opposite to that in which it was moved when the helmsman turned the wheel. If the helmsman stops turning the wheel, hub *12* returns lever *15* to its initial position, the starting valve returns to its central position and the steering engine stops as does the rudder.

If the helmsman turns the wheel at a uniform rate of speed, the axial speeds transmitted to hub *12* by the helmsman and the engine will be equal in magnitude but opposite in direction. Therefore, the starting valve, shifted from its central position by the initial movement of the steering wheel, rests in one position and the engine shaft continues to rotate at the same speed.

When the helmsman starts to turn the wheel faster, thereby transmitting a higher axial speed to the hub *12* than is transmitted by the engine, the shaft of the latter begins to rotate faster, since the starting valve opens the admission ports to a greater degree, and the rudder is put over faster.

On the other hand, if the helmsman turns the wheel slower, thereby transmitting a lower axial speed to the hub *12* than does the engine, the starting valve is shifted to close the ports to a greater degree, the shaft runs slower and the rudder is put over slower.

Of somewhat different design is the servomotor illustrated in Fig. 150. A rod-type transmission *1* from the steering wheel rotates bevel gear *2* which meshes with the large bevel gear *3* keyed on sleeve *4*. Bevel gear *5*, secured to gear *3*, meshes with planetary gears *6* and *7*. The latter mesh with gear *9* mounted on the shaft of the worm wheel *10*. Planetary gears *6* and *7* are freely mounted on pin *8*. The latter is secured to shaft *12* which passes through sleeve *4*. A single-arm lever *13*, mounted at the end of shaft *12*, actuates the differential (starting) valve *14*. Upon turning the steering wheel, transmission *1* and gears *2*, *3* and *5* begin to rotate. Planetary gears *6* and *7*, driven by gear *5*, rotate about their axes and also roll around the stationary gear *9*, thereby rotating shaft *12*. Shaft *12* turns lever *13* and shifts the starting valve *14* from its central position. This starts the steering engine which rotates worm *15*, worm wheel *10*, bevel gear *9* and shaft *11* on which the tiller chain sprocket is mounted. The action of the sprocket begins to put over the rudder.

Rotation of shaft *11* and bevel gear *9* in the direction opposite to that of bevel gear *5* returns the starting valve to its central position.

In addition to those considered here, many other designs and arrangements of servomotor mechanisms have been devised.

It follows from the principles of steering engine and servomotor operation that a definite number of steering wheel revolutions corresponds to a definite number of revolutions of the engine shaft and, therefore, to a definite rudder angle.

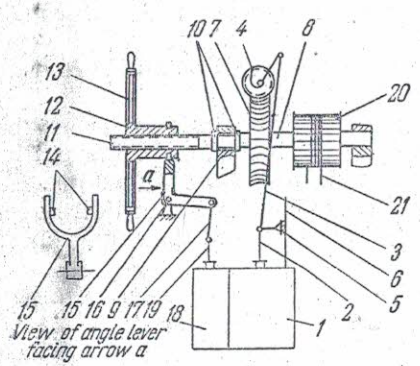

Fig. 149.

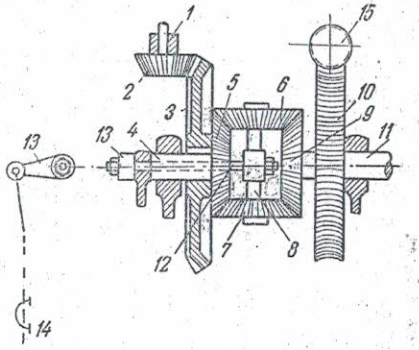

Fig. 150.

The transmission ratio of the servomotor is selected so that the number of steering wheel revolutions required to put the rudder from hard-over to hard-over ranges from 6 to 10.

After mastering the principle upon which the operation of all servomotors is based, there should be no difficulty in understanding the numerous designs employed in practice.

4-3. Construction of Steering Engines

(A) STEAM STEERING ENGINE FOR A DRUM-TYPE STEERING GEAR

Fig. 151 illustrates a steering engine driven by a vertical two-cylinder steam engine, and mounted on base plate *1* and in frame *2*. When the steering engine is to be operated by steam, gear *13*, mounted on a feather on shaft *11*, is shifted to the right by means of lever *16* to disengage it from gear *14*. Besides this, wheel *26* for hand steering with its clutch member *38* should be shifted to the left to disengage it from the clutch member *39* of gear *25*, mounted on the long hollow hub of gear *14*. The wheel is fixed axially in its two positions by the locking device *32*. When this has been done, rotation of the steering

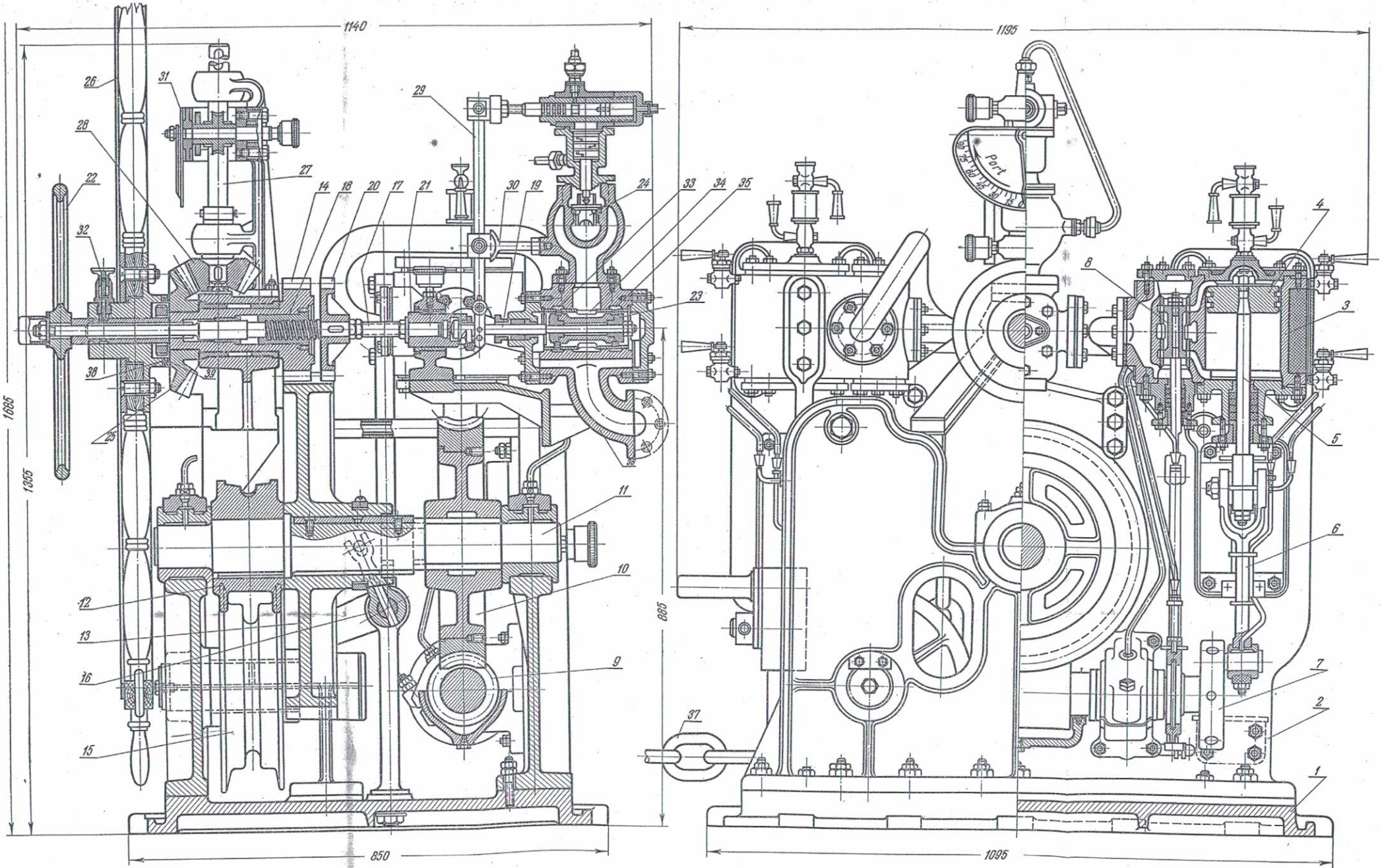
1140
1195
1685
1355
885
850
1095
Port
26
31
28
29
27
22
14
18
20
17
21
30
19
24
33
34
35
32
23
4
8
3
38
39
25
5
11
6
10
12
13
16
9
7
2
37
15
1

Fig. 151

wheel on the bridge is transmitted through a teledynamic system to shaft *27* and gears *28*, *25* and *14*. Since gear *20*, keyed on shaft *17*, meshes with gear *13* which at the given moment is stationary, shaft *17* does not rotate but moves axially due to the threads in the bushing *18* which is fitted and secured in the hub of gear *14*. Bushing *21*, mounted on the end of shaft *17*, moves in the same direction axially. The end of the stem *19* of the starting valve is mounted freely in bushing *21*. Thus the starting piston valve performs only axial movement even if the bushing *21* rotates. Axial movement of bushing *21* opens the economic valve *24* by turning lever *29* about its pivot *30*, and shifts the starting piston valve *23* from its central position by actuating stem *19*.

Live steam, admitted under valve *24*, enters the space *34* between the ends of the starting valve. Let us suppose that piston valve *23* has been shifted to the right. In this case, live steam passes through port *35* to the steam-distribution valves *8* and further to the corresponding ends of the cylinders *3* of the steam engine, actuating its pistons *4*. Spent steam is exhausted through the steam-distribution valves and port *33* into the exhaust connection *36*. By means of piston rods *5* and connecting rods *6*, linked to cranks *7*, power is transmitted to rotate the worm shaft and worm *9* which meshes with worm wheel *10*. Worm wheel *10* is freely mounted on shaft *11* and its hub has clutch jaws of the same type as on the right end of the hub of gear *13*. Thus, when gear *13* is shifted to the right by turning lever *16*, the clutch jaws of the gear and wheel become engaged so that shaft *11* also rotates together with the chain sheave *12* which is keyed to this shaft. Rotation of this sheave actuates the tiller chain *37*, running over the guide rollers *15*, which, in turn, actuates the tiller so that the rudder is put over. At the same time, gear *13* rotates gear *20* and shaft *17*. The latter thereby obtains axial movement opposite to that obtained by turning the steering wheel. Here again, we have the principle of the servomotor. The steering engine can be controlled by handwheel *22* which is usually used for repairs, adjustment, warming up and test runs of the steering engine from the place where it is installed. In addition to power steering described above, provision is made for hand steering either from the bridge or from the steering-engine room.

Hand steering from the bridge is accomplished by disengaging gear *13* from worm wheel *10* and shifting it into mesh with gear *14* by turning lever *16*. In this case, rotation of the steering wheel on the bridge is transmitted to shaft *11* and chain sheave *12* through shaft *27*, and gears *28*, *25*, *14* and *13*.

The rudder can be put over by hand from the steering wheel *26* by releasing locking device *32* and shifting the wheel to the right to engage the clutch jaws *38* and *39* on the end of the wheel hub and on

gear *25*. Then rotation of wheel *26* is transmitted to the chain sheave *12* through gears *14* and *13*.

The steering engine has a rudder indicator, whose hand *31* is actuated through a gear mounted on shaft *27*, and an economic valve which is shown in Fig. 152.

When the steering wheel is at rest, live steam, admitted under valve disk *5*, passes through passage *23*, the open valve in body *22*

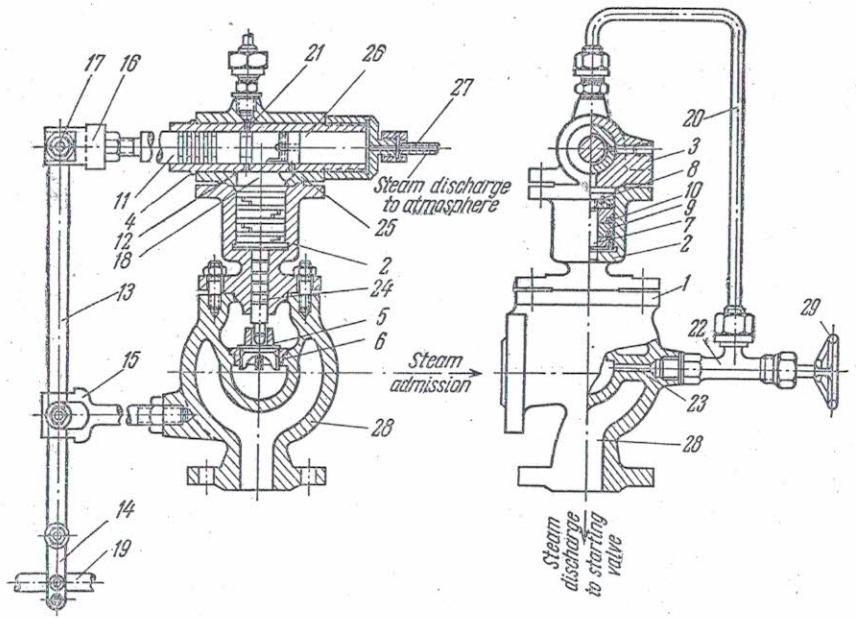

Fig. 152.

and pipe *20* to the port *21* in sleeve *4* of the piston valve *11* which is housed in body *3*. Valve disk *5* fits into seat *6* which is arranged in body *1*. The live steam passes further through the annular groove of the piston valve and port *12* to the cavity *18* of cylinder *2*. The pressure of the steam on piston *7* is transmitted through rod *24* to hold valve *5* in its seat *6*. Upon rotation of the steering wheel and axial movement of stem *19* of the starting valve, link *14* turns the two-arm lever *13* about its pivot in bracket *15*. The top end of lever *13* is hinged through pin *17* to coupling *16*. The motion of lever *13* shifts piston valve *11* to the right or left. In either case, ports *21* and *12* are closed and port *25* is connected to cavity *26*.

This cuts off the admission of live steam to cavity *18* and the pressure in the cavity will drop since it is connected through port *25*, cavity *26* and pipe *27* to the atmosphere.

After this, the pressure of the live steam lifts valve *5* and live steam will bypass into cavity *28* and from there to the starting valve. When the steering wheel comes to rest, stem *19* of the starting valve returns piston valve *11* to the position shown in Fig. 152. This disconnects cavity *18* from the atmosphere and it will again be filled with live steam whose pressure on piston *7* will force valve *5* into its seat *6*.

Rod *24* has labyrinth grooves to prevent the leakage of steam. Piston *7* is sealed by rings *9* in whose grooves packing rings *10* are compressed by the cover *8* of the piston. Handwheel *29* is used to cut out (bypass) the economic valve or to adjust the speed with which the valve disk *5* is lifted and returns to its seat.

One disadvantage of steam-operated economic valves is that a certain amount of steam is wasted in actuating the valve.

Steam losses of a steering engine, when it is at rest, range from 90 to 100 kg per hour if no economic valve is provided. These losses can be avoided if a slight steam lap is provided at both ends of the starting valve, but this increases the time that elapses between the turning of the wheel and the starting of the engine.

(B) STEAM STEERING ENGINE FOR A QUADRANT-TYPE STEERING GEAR

The great length of drum-type chain or rope steering gears, the noise they make in operation and the high mechanical losses have led to the use of quadrant-type steering gear on high-and medium-tonnage ships.

An arrangement with a quadrant-type steering gear powered by a steam steering engine is illustrated in Figs. 153 and 154.

Such steering gears are usually located aft in the tiller room and are widely employed on transport ships.

The steam steering engine of this gear is controlled, as usual, by a starting valve which is actuated through some kind of teledynamic transmission.

The tiller *2* is keyed on the head of the rudder stock *1* and is linked to the quadrant *3* through the buffer device *4*. The springs of this device, compressed at all times, reduce the dynamic load on the parts of the steering engine due to impact of waves or ice floe on the rudder blade in heavy seas or ice fields. Bolt *5* can be used to link the tiller directly to the quadrant if the buffer device gets out of order. The quadrant meshes with the main pinion *6* which is made integral with bushing *7*. The latter has clutch jaws on both end faces. Bushing *7* is freely mounted on vertical shaft *8* of the hand steering arrangement and vertical shaft *9* of worm wheel *10*. A clutch mounted on these same vertical shafts consists of handwheel *11*, upper clutch

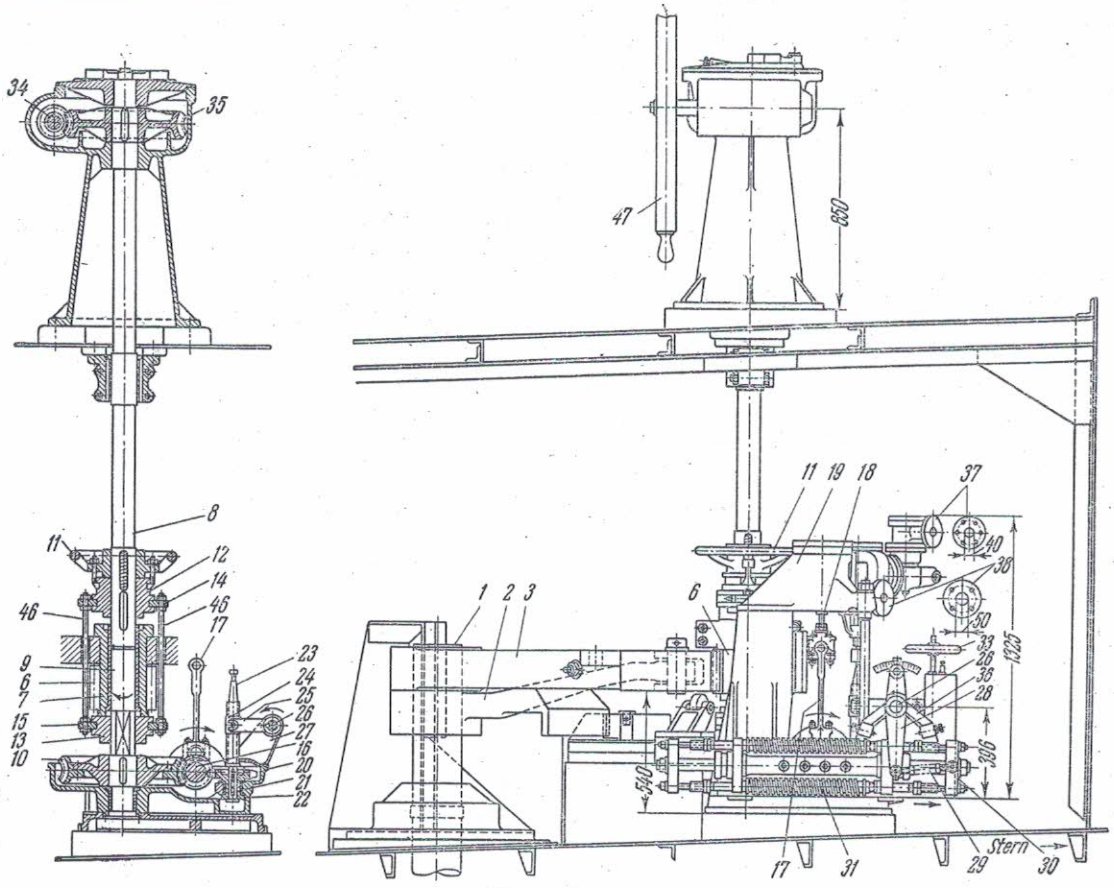

Fig. 153.

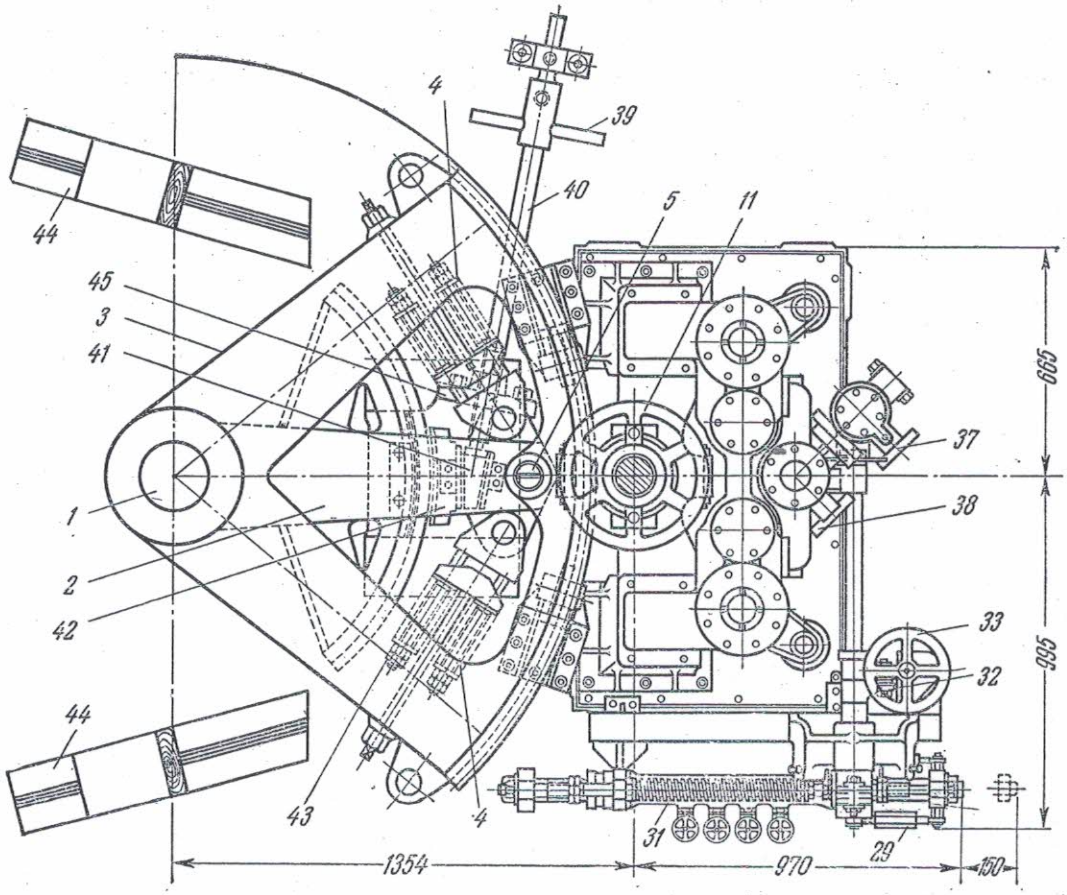

Fig. 154.

member *12* and lower clutch member *13* which are joined by split collars *14* and *15* and tie bolts *46*. Split collars *14* and *15* are linked to the clutch members in such a way that the members are free to rotate. Clutch member *12* can slide along a key on shaft *8* while clutch member *13* is made with a sliding fit on the hexagonal part of shaft *9*. Thus the clutch can be slid up or down by turning handwheel *11*. Worm wheel *10* is keyed on shaft *9* and meshes with worm *16* which is made integral with the crankshaft of the steam engine.

Connecting rods *17* and piston rods *18* link the crankpins of the crankshaft to the pistons of the steam engine cylinders *19*.

Worm *16* meshes with gear *20* whose hub *21* runs freely in bearing *22*. Vertical shaft *23* is mounted in the hub of gear *20* and is linked to it by means of a feather. The threaded part of shaft *23* runs through a nut *24* whose trunnions enter the forked ends of lever *25* which is keyed on the shaft *26*. The latter turns in bearings mounted in brackets *27*. The stem of the starting valve is linked to the upper end of vertical shaft *23* by a coupling designed so that only axial motion is transmitted to the stem. Shaft *26* is linked through lever *28* and a hinged joint to tie rod *29* which, in turn, is linked to crosshead *30* of the hydraulic telemotor receiver *31*. Coupling *32* links shaft *26* with the lever of handwheel *33* for tiller room hand control of the starting valve through receiver *31*. Now we shall consider the operation of the steering engine in power steering from the steam engine and in hand steering.

In power steering the hand steering arrangement must be disengaged by rotating handwheel *11* to put the clutch in the position shown in the illustrations. Moreover, coupling *32* for tiller room control of the starting valve must be disengaged. In this case, rotation of the steering wheel on the bridge is transmitted by the hydraulic telemotor and leads to movement of the receiver crosshead *30*. This actuates tie rod *29* and turns lever *28*. As a result, shaft *23* moves axially and shifts the starting valve from its central position. Lever *25* and shaft *26* will also be turned. In this manner, the steam engine is started. Consequently, the engine crankshaft, worm *16*, gear *20* and shaft *23* begin to rotate.

The rotation of gear *20* and shaft *23* runs the nut *24* along the shaft in the axial direction opposite to that in which it was shifted upon the turning of shaft *26*. Depending upon the ratio of the length of travel of nut *24* along shaft *23* and its axial shift with the shaft, the starting valve will approach or withdraw from its central position.

If the helmsman stops turning the wheel, shaft *26* will stop turning and shaft *23* will screw into or out of nut *24* until it brings the starting valve to its central position. This stops the steam engine and, consequently, rudder movement.

The steam engine can be controlled from handwheel *33* after engaging coupling *32*. Hand steering from wheel *47* is accomplished by lowering clutch member *12* so that it engages the upper end of bushing *7* and clutch member *13* is disengaged from the bushing. If, after this, steering wheel *47* is turned, worm *34*, mounted on the wheel shaft, rotates worm wheel *35* keyed on shaft *8*. Rotation is transmitted by this shaft to clutch member *12*, bushing *7* and, consequently, to main pinion *6* which actuates quadrant *3*, tiller *2* and, finally, rudder stock *1*. Stops *36* are provided to limit the travel of the starting valve. Live steam is supplied to the engine through connection *37*, and spent steam is exhausted through connection *38*.

This steering gear incorporates a friction brake capable of locking the gear in any position. The brake is applied by turning handle *39* of rod *40*, one end of which passes through threaded flange *45*. Tapered pressure block *41*, mounted freely on the same end of rod *40*, is fitted on a sliding dovetail joint with brake shoe *42* and upright *43*, the latter being secured to the foundation frame of the steering gear.

Rotation of rod 40 displaces the pressure block *41* along the axis of the rod and shifts the brake shoe *42* along the ship's middle-line plane. This applies the brake and the rudder can be locked at any angle.

Stops *44* limit rotation of the quadrant and thus prevent the rudder from being put over too far and damaged.

(C) ELECTRIC QUADRANT-TYPE STEERING GEAR

An electric steering gear is usually installed on ships having electrically operated auxiliary machinery.

An electric quadrant-type steering gear with an additional hand steering arrangement is shown in Fig. 155.

The hub *1* of the tiller is keyed on the rudder stock *2*. The tiller is linked through buffer springs to quadrant *6* which meshes with main pinion *7* keyed on shaft *8*. Bushing *9* is secured on keys on the same shaft while hub *21* of worm wheel *10* is freely mounted. The latter meshes with worm *11*. Splines *14* are provided on the outside surface of bushing *9* and splines *16* on the inner surface of housing *12* which is the upper part of worm wheel *10*. Friction disks *15* and *13* with slots on their inner and outer circumferences, respectively, are fitted on these splines in alternate order comprising a friction clutch for engaging or disengaging the worm wheel and shaft *8*.

Power steering, in which worm *11* is driven by an electric motor, is effected by turning handwheel *22* to compress disk springs *23*. This applies pressure to the friction disks which engage worm wheel *10* with shaft *8* and, consequently, link it to the quadrant through

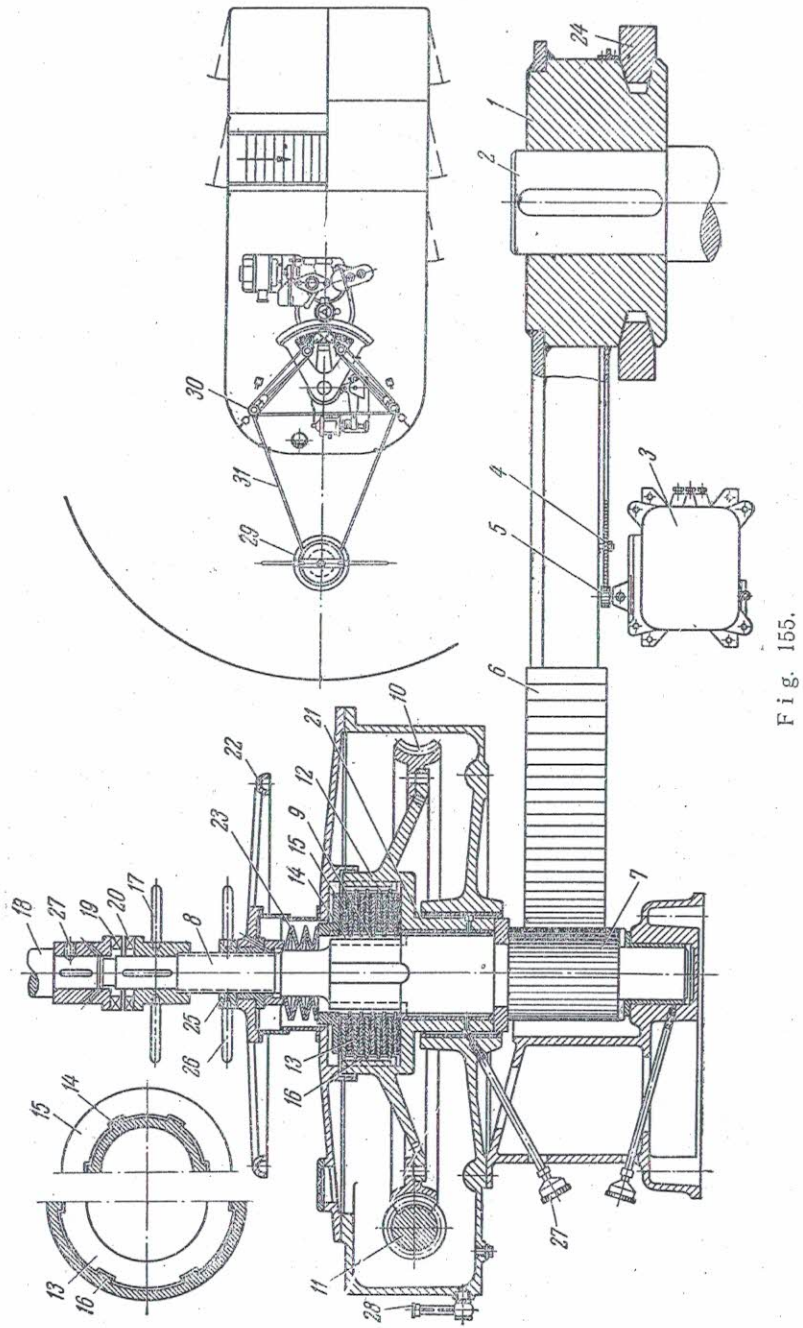

Fig. 155.

pinion 7. Clutch members *19* and *20* should be in the position shown, i. e., disengaged.

If hand steering is required, the friction clutch is disengaged, clutch member *20* is turned to the position for engagement with handwheel *17*, stop screw *27* is released and clutch member *19* is shifted down into engagement with member *20*. After this, rotation of shaft *18* is transmitted to shaft *8* and pinion *7* which turns quadrant *6* and rudder stock *2*.

The rudder stock is locked in a definite position by applying brake *24*. The steering gear is stopped at the maximum permissible rudder angle by limit switch *3* which is actuated by segment gear *4* and pinion *5*. Hub *25* of handwheel *26* is used to lock handwheel *22*. The bearings are lubricated through grease cups *27*. The worm gearing operates in an oil bath whose level is checked by means of the sight oil gauge *28*.

Rudder tackle *30* is sometimes used as standby gear for emergencies. It is powered from the stern capstan 29 through wire rope *31*.

(D) HYDRAULIC STEERING GEARS

Hydraulic steering gears are distinguished for their compact arrangement, silent operation, small overall size in conjunction with a high torque, smooth action, dependability and ability to operate under various climatic conditions.

Hydraulic steering gears may be either of the ram or vane type, which may be either power or hand operated. The ram type may have one, two or four rams.

Hand hydraulic gear is installed on low-tonnage self-propelled vessels and nonpropelled barges, and also serve as standby drive for power-driven hydraulic steering gear. The latter are installed on medium- and high-tonnage ships.

Hydraulic steering gears are operated by rotary pumps with either hand or power drives. The pumps may have either constant or variable delivery; they may or may not be reversible.

The two-ram hydraulic steering gear shown in Fig. 156 is powered by a variable-delivery nonreversible rotary radial-piston pump and is controlled by a telemotor or a starting motor which actuates tie rod *11*.

Let us suppose that drive motor *26* of pump *7* is switched on and rotates the pump shaft on which cylinder body *1* is mounted, and that the steering wheel is being held stationary. In this case, the floating ring *4* (central figure) is concentric with the casing *7*. The slippers *3* slide along the inside surface of the floating ring *4* while the pistons *2*, rotating together with cylinder body *1*, are stationary in their cylinders, i. e., no pumping action takes place.

If the helmsman shifts tie rod *11* to the left by turning the steering wheel, through joint *12* the rod will turn lever *28* in the clockwise direction about pivot *14*. Through joint *13*, lever *28* shifts spindle *5*, and with it floating ring *4*, to the left (figure at left). As a result, the cylinders below the stationary partition *8* begin to draw the liquid from ram cylinder *18* through the piping and port *10*. At the

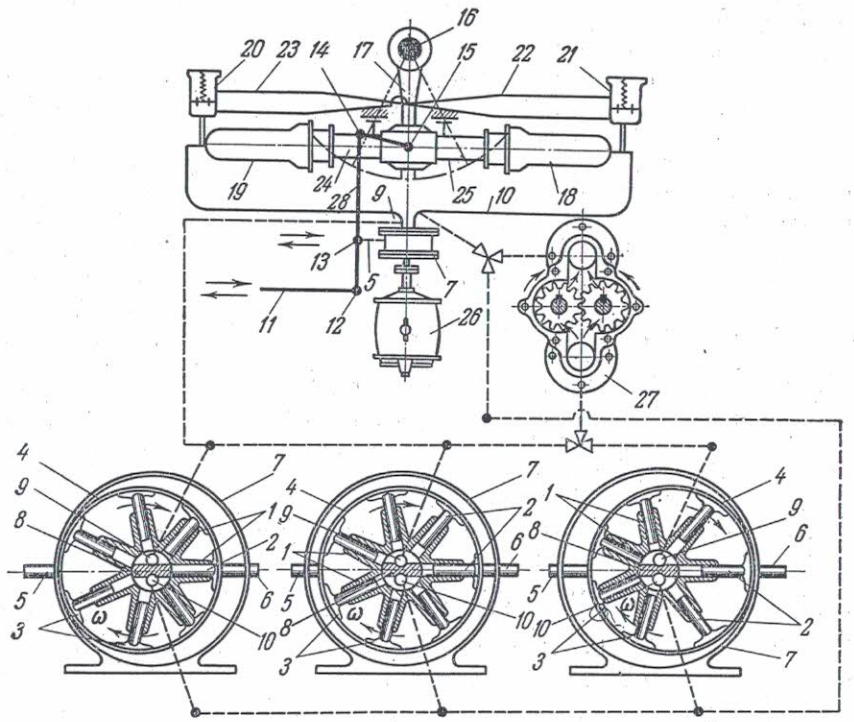

Fig. 156.

same time, the pistons of the cylinders above partition *8* begin to discharge the liquid through port and piping *9* into ram cylinder *19*. Thus, rams *24* and *25* will move to the right, turning the tiller *17* and rudder stock *16* in the counterclockwise direction. Lever *28* has a floating action and also moves to the right with the rams since it is linked to the swivel crosshead of the tiller by joint *15*. This action tends to return floating ring *4* back to its central position. Thus the servomotor principle, or hunting method, brings the pump to its "no delivery" position when the desired rudder angle has been obtained. If the floating ring is shifted to the right by the action of

the steering wheel (figure at right) suction occurs in the cylinders above partition *8* and liquid is drawn from ram cylinder *19* through piping and port *9*. The cylinders below the partition discharge liquid through piping and port *10* into ram cylinder *18*. Now, tiller *17* and rudder stock *16* turn clockwise. Valves *20* and *21* serve to pass liquid from pipeline *22* to *23*, or vice versa, if the pressure in one of them becomes excessive. The pump discharge and, consequently, the speed of rudder motion depend upon the amount the floating ring is shifted from its central position. The steering gear can also be operated from the reversible standby gear pump *27*.

In recent years, hydraulic steering gears powered by constant-delivery, nonreversible pumps are used more and more on ships. This is due to their simpler design and maintenance in comparison with the variable-delivery types.

Steering gear can be controlled by a hydraulic telemotor of any design. The hydraulic steering gear with the constant-delivery screw pump shown in Fig. 157 is equipped with a telemotor having a slanted thrust ring. The whole telemotor mechanism is housed in the wheel pillar *1*. Turning together with the steering wheel *3* on shaft *8* is the cylinder body *2* which has a series of axial cylinder bores around its circumference. Pistons *4*, forced apart by springs *5*, are fitted into the cylinder bores. The ends of the pistons bear against slanted thrust rings designed in the form of ball thrust bearings *6* mounted on the covers *7*.

Let us suppose that the hydraulic steering gear is ready for operation, that the drive motor of the nonreversible constant-delivery pump is switched on and the pump delivers liquid along pipeline *33* to the valve chest *55*. Since piston valve *28* is in the central position, the liquid passes through the clearances between the valve and the chest and is drawn into pump *48* again. Thus, the working fluid continuously circulates and the rudder is at rest.

Now, if the steering wheel is turned clockwise, the pistons in the cylinder body to the right of a vertical axis will approach each other in the bores, compressing the springs, and displace the liquid between the pistons, forcing it out through the radial passages *13* into the cavity and passage *9*. From here the liquid will be delivered through pipeline *14*, open stop valve *16*, pipeline *21*, valve box *24* of the telemotor and pipeline *25* into the right end of servomotor *26* where it shifts piston *23* to the left. At the same time, liquid from the left end of servomotor *26* is discharged through pipeline *27*, left-hand passages of valve chest *24*, pipeline *22*, the open nonreturn valve *17* and pipeline *15* into cavity *12* from where it is drawn through passage *10* and the radial passages *13* into the cylinder bores to the left of the vertical axis, in which the pistons are spread apart by the springs.

As piston *23* of servomotor *26* moves to the left it shifts lever *46* of distribution valve *28* in the same direction until the pistons *29* and *30* of this valve run up against the shoulders *31* and *32*. This stops the circulation of the liquid that takes place when the distribution valve *28* is in its central position. Now the liquid delivered by pump *48* passes along pipeline *33*, through the left end *34* of valve chest *55* and pipeline *53* into cavity *72* of valve chest *36*. After lifting valve *52* the liquid passes into cavity *73* and further through pipeline *49* into the right ram cylinder *47*. At the same time, liquid from cavity *72* in valve chest *36* passes through passage *50* under piston *51* of valve *38*, lifting this valve. As a result, pump *48* draws in liquid from the left ram cylinder *41* through pipeline *40*, cavity *39*, valve *38*, pipeline *35*, right end of valve chest *55* and pipeline *54*. Rams *42* and *43* move to the left and turn the rudder *44* clockwise. In turning, the tiller pushes tie rod *45* to the left and, by means of lever *46*, piston *23* of servomotor *26* is returned to its central position. Piston *23* can be shifted from its central position by turning handwheel *56* to rotate pinion *57* which actuates rack *58* and, through it, the rod of the piston. When the steering wheel is turned in the opposite direction, the rudder stock is turned counterclockwise, as can easily be ascertained if the interaction of the parts is followed out for this case.

In case the power drive of the steering gear gets out of order, i. e., the rams fail to move, piston *23* of the servomotor will shift to one of its extreme positions. The pressure in the telemotor system will increase and valve *59* or *60* (depending upon the position of the piston) is shifted to connect pipelines *22* and *61* or *21* and *62*, respectively, running to ram cylinders *41* and *47*. In this manner, automatic changeover to hand operation of the steering gear is accomplished, and the telemotor serves as a hand pump. If required, the rudder can be put over by a hand drive through pinion *63*, meshing with quadrant *64*.

The swivel crosshead *65* of tiller *44* has slide blocks *66* which slide along guides *67* fastened to the base plate *68*. The transmitter *69* is coupled to the tiller and indicates the rudder position on an indicator on the bridge. Oil is poured into the wheel pillar of the telemotor through screen *70*; the oil level is measured by means of the gauge stick *71* and air is purged from the system through vent plug *20*. Nonreturn valves *18* and *19* serve to fill the telemotor system with oil from the wheel pillar. The oil level in the telemotor should never drop below the lower mark on the gauge stick. Oil is bypassed from one ram cylinder to the other in case of a high dynamic load on the rudder (for instance, impacts from waves) through relief valve *76*.

In emergency hand steering with quadrant *64*, valve *74* should be lifted by turning its threaded spindle *75*.

Vane-type hydraulic steering gears have been devised for ships with the intention of reducing the weight and overall size of such machinery and simplifying their design.

One arrangement of a vane-type gear using a nonreversible, constant-delivery pump is shown schematically in Fig. 158. The pump of the telemotor, actuated by the steering wheel *2*, is housed in the wheel pillar *1*. Any tube of pump may be employed. When the steering wheel is at rest, valve *5* will be in its central position and the liquid delivered by pump *9* through pipeline *10* into cavity *11* of the valve chest returns to the pump through cavities *12* and *13* and pipeline *14*.

Let us suppose that the liquid delivered by the telemotor pump along pipeline *3* enters cavity *4* in valve chest *6* of control valve *5*. At the same time, the telemotor pump will draw liquid from cavity *7* of the valve chest through pipeline *8*, thereby shifting valve *5* to the right. Liquid, delivered by pump *9* along pipeline *10* into cavity *11*, enters cavity *16* through cavity *12* and passage *15*, and shifts valve *17* to the left. From cavity *16* the liquid is admitted through pipeline *18* into chambers *19* and *20* of the rotary hydraulic actuating drive *21*. From chambers *22* and *23* of this drive, liquid is drawn through pipeline *24*, valve chest *25*, passage *26*, cavity *27* and pipeline *14* into pump *9*. The difference in the pressure in chambers *19* and *20* and the pressure in chambers *22* and *23* is applied to the vanes *28* which begin to rotate clockwise, turning the rudder stock *29* in the same direction. When the helmsman stops turning the wheel, the pressure in pipeline *3* drops, springs *30* and *31* return valves *5* and *17* to their central positions and the vanes *28* and rudder stock *29* come to rest. At high dynamic loads on the rudder, exceeding the permissible values, the relief-bypass valve *33* opens and bypasses liquid from one set of chambers of the actuating drive to the other set.

Buffer devices are provided in the form of balls *34*, mounted in the vanes, which run against strikers *35* in the partitions *36* at the extreme positions of the vanes. Hand steering is accomplished by means of the telemotor pump. Turning the handle *32* of the three-way cocks connects the telemotor pump to pipelines *37* and *38*, and disconnects it from pipelines *3* and *8*.

If the telemotor pump delivers liquid through pipeline *37*, the liquid will pass into cavity *39* of valve chest *40* and shift piston valve *41* to the right. The liquid will pass further along pipeline *42* into chambers *19* and *20* of the actuating drive *21*. At the same time the pump will draw liquid from chambers *23* and *22* through pipeline *43*, valve chest *40* and pipeline *38*. As in the previous case, the

difference in pressures in the chambers will cause the vanes to turn clockwise.

As soon as the helmsman stops turning the wheel the pressure in the system drops, valve *41* is returned to its central position by spring *44* and the rudder comes to rest.

In cases when the rudder is operated by emergency steering facilities (quadrants, rudder tackle, etc.), compression of the liquid in the chambers is prevented by opening the relief-bypass valve *33* by its spindle *45*.

The interaction of the parts of this steering gear for counter-clockwise rotation of the rudder can be followed out in Fig. 158.

4-4. Determining the Principal Data Required in the Design of Steam and Electric Steering Gears

The main initial data required to determine the principal dimensions of steering gears are the rudder characteristic, χ_r, the torque, M_{rs}, in kg-m developed on the rudder head and the time, τ, required to put over the rudder.

The time required to put the rudder from hard-over to hard-over, depending upon the purpose of the ship and used in steering gear design, is listed in Table 47. It should not exceed the standards established by the U. S. S. R. Shipping Register.

The time that elapses before the steering engine reaches its rated speed, which we shall call the starting time, must be taken into consideration by reducing the time τ for putting the rudder from hard-over to hard-over by 1.5 to 2 seconds.

If we denote the gearing ratio between the rudder stock and steering engine shaft as i_{sg}, the overall efficiency of the steering gear as η_{sg} and the speed at which the rudder stock turns,

Table 47

Type of ship	Time required to put rudder from hard-over to hard-over, sec	Speed of rudder movement, deg/sec, for rudder angle of	
		$2\alpha°=70°$	$2\alpha°=64°$
Ice breakers	15	4.66	4.25
Sea-going craft and transport ships . . .	25 to 30	2.8 to 2.34	2.56 to 2.13
Towboats	20 to 25	3.5 to 2.8	3.2 to 2.56
River craft	40 to 45	1.75 to 1.55	1.6 to 1.44

expressed in rpm, as n_{rs}, then the torque developed on the steering engine shaft and its speed, n_m rpm, will be

$$M_m = \frac{M_{rs}}{i_{sg}\eta_{sg}} \text{ kq-m} \quad (312)$$

$$n_m = i_{sg} n_{rs} \text{ rpm} \quad (313)$$

where n_m = 100 to 350 rpm for steam engines
n_m = 300 to 1,800 rpm for electric motors.

The angular velocity of rotation ω_{rs} of the rudder stock can be calculated from the following formulas:

$$\omega_{rs} = \frac{\pi n_{rs}}{30} \text{ 1/sec} \quad (314)$$

$$\omega_{rs} = \frac{2\alpha°}{\tau}\frac{\pi}{180°} \text{ 1/sec} \quad (315)$$

where $\alpha°$ = maximum rudder angle from the middle-line plane.

It follows from formula (314) that

$$n_{rs} = \frac{30\omega_{rs}}{\pi} \text{ rpm} \quad (316)$$

Combining equations (315) and (316) we obtain

$$n_{rs} = \frac{30}{\pi}\frac{2\alpha°}{\tau}\frac{\pi}{180} = \frac{1}{3}\frac{\alpha°}{\tau} \text{ rpm} \quad (317)$$

Combining equations (313) and (317) we can write

$$i_{sg} = \frac{n_m}{n_{rs}} = \frac{n_m}{\frac{1}{3}\frac{\alpha°}{\tau}} = 3n_m\frac{\tau}{\alpha°} \quad (318)$$

Taking equations (314) and (315) into consideration, the power developed on the rudder stock is

$$N_{rs} = \frac{M_{rs}\omega_{rs}}{75} = \frac{M_{rs}}{75}\frac{2\alpha°}{\tau}\frac{\pi}{180°} = 4.65\frac{M_{rs}}{10^4}\frac{\alpha°}{\tau} \text{ metric hp} \quad (319)$$

$$N_{rs} = \frac{M_{rs}\omega_{rs}}{75} = \frac{M_{rs}}{75}\frac{\pi n_{rs}}{30} = 1.395\frac{M_{rs}n_{rs}}{10^3} \approx 1.4\frac{M_{rs}n_{rs}}{10^3} \text{ metric hp} \quad (320)$$

The shaft horse power of the steering engine motive unit will be

$$N_m = \frac{N_{rs}}{\eta_{sg}} = 4.65\frac{M_{rs}}{10^4\eta_{sg}}\frac{\alpha°}{\tau} \text{ metric hp} \quad (321)$$

$$N_m = \frac{N_{rs}}{\eta_{sg}} = 1.4\frac{M_{rs}}{10^3\eta_{sg}}n_{rs} \text{ metric hp} \quad (322)$$

The shaft horse power can also be determined from the shaft torque

of the motive unit. Thus

$$M_m = \frac{M_{rs}}{i_{sg}\eta_{sg}} = \frac{M_{rs}}{\frac{3\tau n_m}{\alpha^\circ}\eta_{sg}} = \frac{M_{rs}}{\eta_{sg} n_m}\frac{1}{3}\frac{\alpha^\circ}{\tau} \text{ kg-m} \quad (323)$$

$$N_m = \frac{M_m n_m}{716.20} = \frac{M_{rs} n_m}{716.20\eta_{sg} n_m}\frac{\alpha^\circ}{3\tau} = \frac{M_{rs}}{716.20\eta_{sg}}\frac{\alpha^\circ}{3\tau} = \frac{M_{rs}}{2148.6\eta_{sg}}\frac{\alpha^\circ}{\tau} \text{ metric hp} \quad (324)$$

The design of a steering gear with a power drive is based upon the assigned shaft speed of the motive unit, while for hand steering the required speed of the steering wheel is calculated from the assigned peripheral speed, v_{sw}, of the wheel rungs, using the equation:

$$60 v_{sw} = \pi D_{sw} n_{sw}$$

$$n_{sw} = \frac{60 v_{sw}}{\pi D_{sw}} \text{ rpm} \quad (325)$$

where $v_{sw} \approx 0.6$ to 0.8 m per sec for the main hand steering gear
$v_{sw} \approx 0.6$ to 1 m per sec for the standby hand steering gear
$D_{sw} \approx 0.75$ to 2 = diameter of the steering wheel, m.

The angular velocity of the steering wheel is

$$\omega_{sw} = \frac{\pi n_{sw}}{30} \quad (326)$$

Thus, the gearing ratio for a hand steering gear will be

$$i_{hs} = \frac{n_{sw}}{n_{rs}} = \frac{n_{sw}}{\frac{1}{3}\frac{\alpha^\circ}{\tau}} = 3 n_{sw}\frac{\tau}{\alpha^\circ} \quad (327)$$

Denoting the efficiency of the hand steering gear as η_{hs}, the torque that must be applied to the steering wheel shaft in hand steering will be

$$M_{sw} = \frac{M_{rs}}{i_{hs}\eta_{hs}} \quad (328)$$

The force that must be applied to the steering wheel rungs is

$$T_{sw} = \frac{M_{sw}}{\frac{D_{sw}}{2}} = \frac{2M_{sw}}{D_{sw}} \quad (329)$$

where $D_{sw} = 0.75$ to 2 m = diameter of a circle described by the rungs of the steering wheel.

The force that must be exerted by each helmsman is

$$p_{hm} = \frac{T_{sw}}{k_{hm}} \quad (330)$$

where k_{hm} = the number of helmsmen.

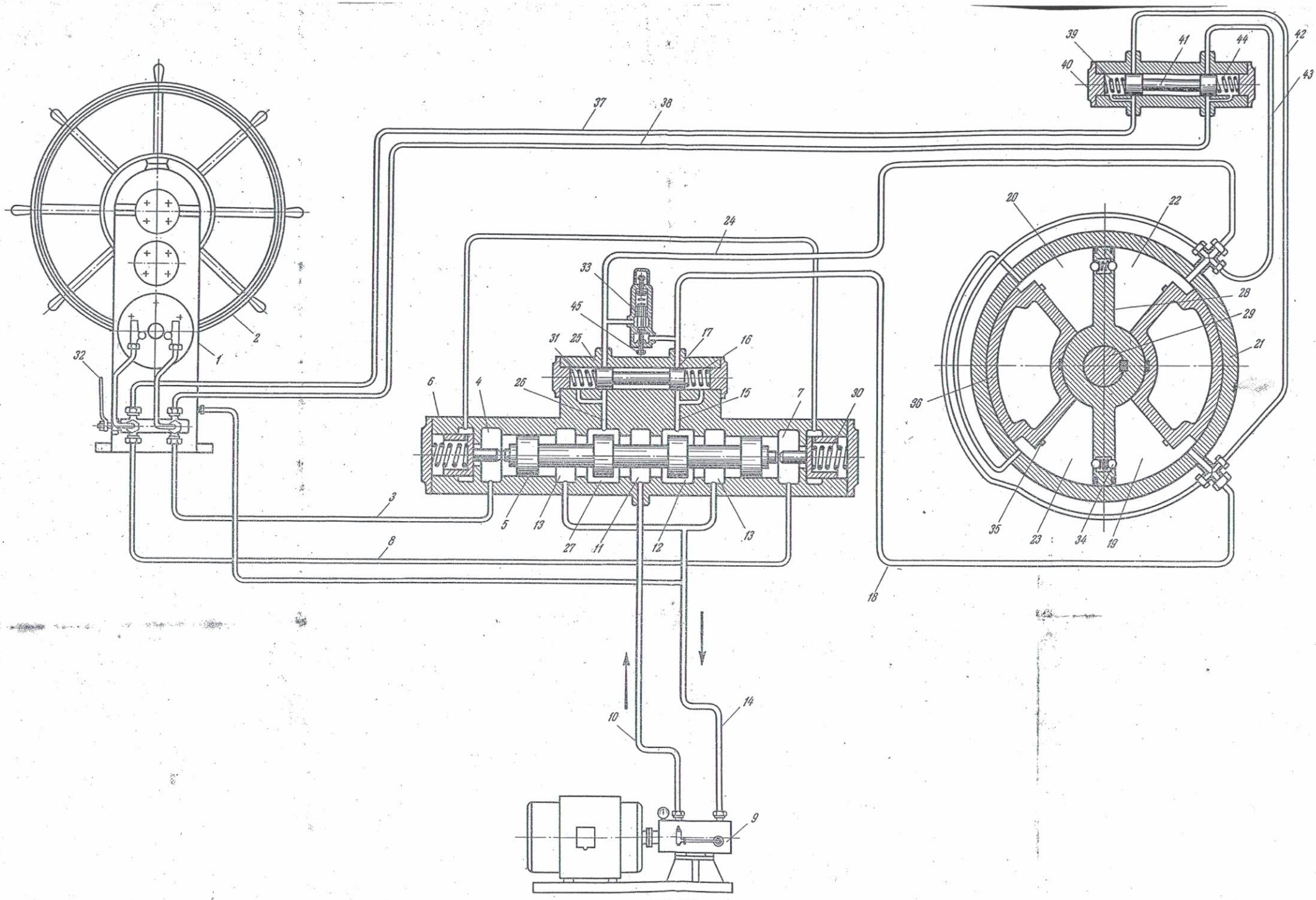
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Fig. 158

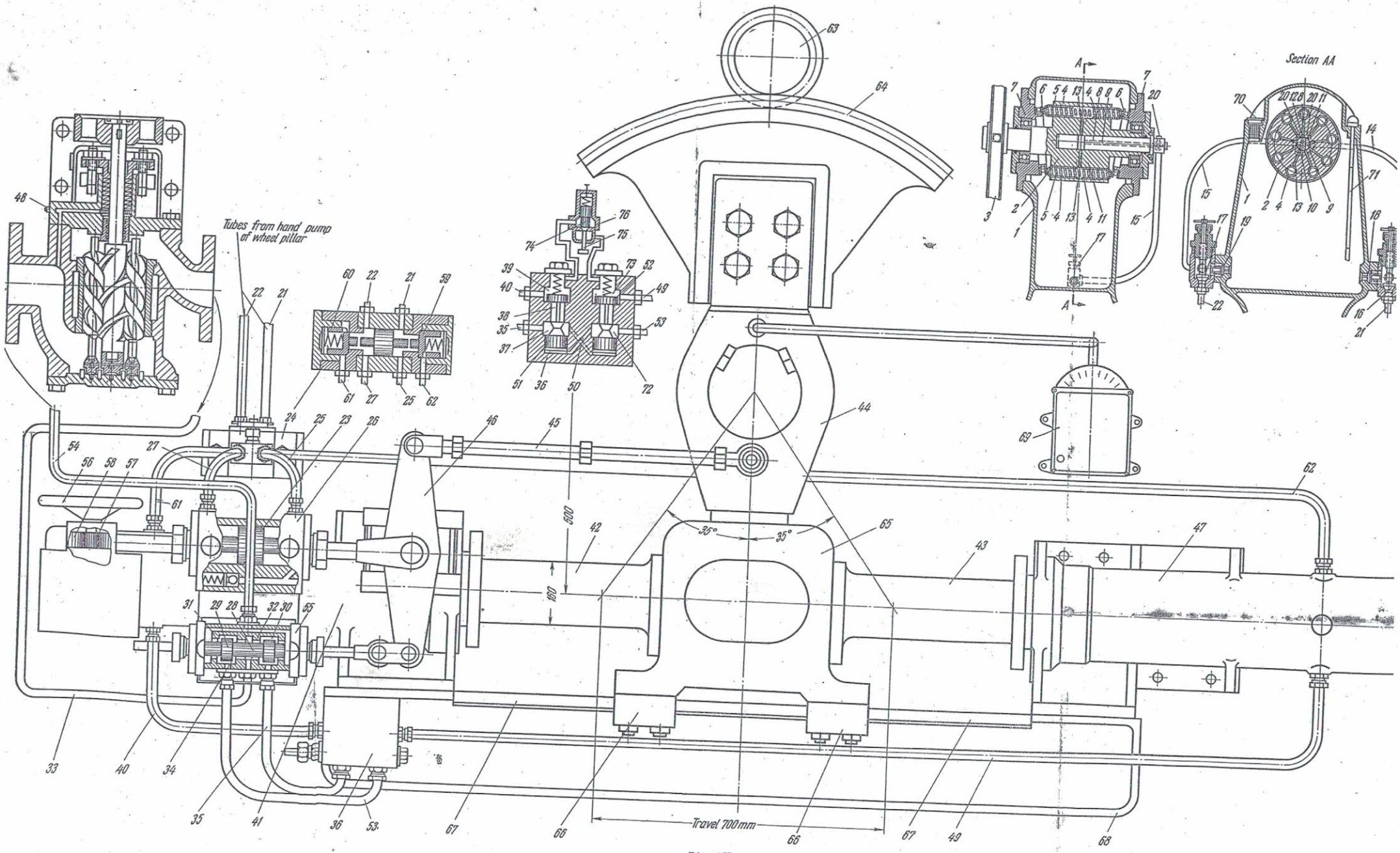

Tubes from hand pump of wheel pillar
Section AA
Travel 700 mm
35°
35°
600
180

Fig. 157

Table 48

α^c	τ, sec	$n_{rs} = \frac{1}{3}\frac{\alpha^\circ}{\tau}$, rpm	$\omega_{rs} = \frac{\pi n_{rs}}{30}$, 1/sec	$i_{sg} = 3n_m\frac{\tau}{\alpha^\circ}$ $i_{hs} = 3n_{sw}\frac{\tau}{\alpha^\circ}$	$n_{sw} = \frac{60\,v_{sw}}{3.14D_{sw}}$ rpm				$i_{hs} = 3n_{sw}\frac{\tau}{\alpha^\circ}$		$i_{hs} = \frac{M_{rs}}{k_{hm}p_{hm}D_{sw}\eta_{hs}}$	
					for		for		for		for	
					$v_{sw} = 0.8$ m/sec	$v_{sw} = 1$ m/sec	$D_{sw} = 0.75$ m	$D_{sw} = 2$ m	$D_{sw} = 0.75$ m	$D_{sw} = 2$ m	$D_{sw} = 0.75$ m	$D_{sw} = 2$ m
max		min	min	max	max		min		max		min	
For power steering gears												
32	30	0.355	0.036	$2.84n_m$	—	—	—	—	—	—	—	—
For standby hand steering gears												
20	60	0.111	0.111	$9n_{sw}$	—	$\frac{19.1}{D_{sw}}$	25.5	9.5	229.5	81.5	$\frac{M_{rs}}{41\eta_{hs}}$	$\frac{M_{rs}}{128\eta_{hs}}$
For main hand steering gears												
32	30	0.355	0.036	$2.84n_{sw}$	$\frac{15.3}{D_{sw}}$	—	20.4	7.65	5.8	21.8	$\frac{M_{rs}}{9\eta_{hs}}$	$\frac{M_{rs}}{24\eta_{hs}}$

The tabular values disregard the acceleration of the motive unit of the steering engine for hydrofoil rudders.

Equation (327) determines the gearing ratio on the basis of the speed with which the steering wheel is turned. Therefore, the effort on the rungs or the number of helmsmen required may not conform with the stipulations of the Shipping Register. In this case, the gearing ratio $i_{hs} = \frac{\omega_{sw}}{\omega_{rs}}$ should be determined from the following equations:

$$M_{rs}\omega_{rs} = M_{sw}\omega_{sw}\eta_{hs} = \frac{T_{sw}D_{sw}}{2}\,\omega_{sw}\eta_{hs} = \frac{p_{hm}k_{hm}D_{sw}}{2}\,\omega_{sw}\eta_{hs}$$

$$i_{hs} = \frac{\omega_{sw}}{\omega_{rs}} = \frac{2M_{rs}}{p_{hm}k_{hm}D_{sw}\eta_{hs}} \qquad (331)$$

Thus the gearing ratio should not exceed the value obtained in equation (327) or be less than that obtained in equation (331).

Table 48 has been compiled on the basis of the limiting values stipulated in the U.S.S.R. Shipping Register for steering gears.

Approximate values of η_{sg} and i_{sg} for steering gears with various types of steering engines are given in Table 49.

Table 49

Steering gear with	Gearing ratio, i_{sg}	Overall efficiency of steering gear, η_{sg}
Steam steering engine	284 to 1,105	0.1 to 0.35
Electric steering engine	1,105 to 5,000	0.1 to 0.35
Main hand steering gear	4 to 22	0.36 to 0.7
Standby hand steering gear	4 to 229	0.36 to 0.4

The gearing ratio of a steering gear with a steam steering engine, as well as with an electric drive or that of hand steering gears, may exceed the limits indicated in Table 49 and may depend upon the design of the steering engine and upon the purpose and size of the ship.

The overall efficiency of the steering gear depends upon the type of kinematic linkage between the rudder stock and steering engine shaft and is equal to the product of the efficiencies of the elements of the gear train used and of the shaft bearings.

The following efficiency values may be taken to determine the overall efficiency:

Quadrant meshing with pinion, pair of meshing spur gears	$\eta_{pg} = 0.93$ to $0.96 = \eta_{qp}$
Vertical shaft running in plain bearings	$\eta_{vs} \approx 0.94$
Horizontal shaft running in plain bearings	$\eta_{hs} \approx 0.97$
Shaft running in antifriction bearings	$\eta_{sh} \approx 0.99$
Hinged couplings for linking shafts	$\eta_{hc} = 0.94$ to 0.98
Toothed or rope quadrant with the tiller	$\eta_{qt} = 0.90$ to 0.94
Chain drum on axle or shaft	$\eta_{cd} = 0.88$ to 0.92
Chain sprocket on axle or shaft	$\eta_{cs} = 0.92$ to 0.94
Pocketed sheave on axle or shaft	$\eta_{ps} = 0.92$ to 0.94

The efficiency of worm gearing, when the transmission is from worm to worm wheel, is

$$\eta_{wg} = \eta_{wb}\frac{\tan\beta_p^\circ}{\tan(\beta_p^\circ + \varphi_f^\circ)} \qquad (332)$$

or, when the transmission is from the worm wheel to the worm,

$$\eta_{wg} = \eta_{wb}\frac{\tan(\beta_p^\circ + \varphi_f^\circ)}{\tan\beta_p^\circ} \qquad (333)$$

where β_p° = helix angle of the worm at the pitch diameter, depending upon the number of threads, or starts (Table 50)

$\varphi_f = 5^\circ$ to 8° = angle of friction

η_{wb} = efficiency of the worm shaft bearings.

It proves expedient to select single-thread worms so as to increase the running speed of steering engines since this reduces their overall size and weight. Worm gearing efficiency will however be lower than in the case of two- or three-thread worms as the efficiency increases with the helix angle of the worm.

If self-braking is one of the necessary features of a steering gear, as is usually the case when an electric motor is used to drive a quadrant- or drum-type steering engine, angle β_p° must be less than the angle of friction. This, of course, reduces the efficiency of the steering engine.

The high friction forces in a steam steering engine and the use of a differential valve and servomotor exclude the need of a self-braking (irreversible) worm for such arrangements. The limiting values of angle β_p° for worms with a standard number of threads are given in Table 50.

Table 50

Type of worm	β_p°
Single-thread	4° to 7°
Double-thread	8° to 14°
Triple-thread	12° to 20°
Quadruple-thread	17° to 26°

Denoting the total area of the rudder, in sq m, as F and the distance from its centre of gravity to its axis of rotation, in m, as r, we can express the rudder characteristic as

$$\chi_r = 100\,Fr \tag{334}$$

If semibalanced or balanced rudders are employed, the characteristic calculated from the preceding equation must not be less than that of the semibalanced or balanced part of the rudder. Thus

$$\chi_r = 100 F_b r_b \tag{335}$$

where F_b = area of the balanced or semibalanced part of the rudder, sq m

r_b = distance from the centre of gravity of the balanced or semibalanced part of the rudder to the axis of rotation, m.

Using the rudder characteristic we can find in Table 51, in accordance with the speed of the ship, the diameter of the part of the rudder stock that is subject to torsional stresses.

Table 51

Characteristic $\chi_r = 100\,Fr$	Maximum speed, knots							
	Not over 8	10	12	14	16	18	20	22
	Rudder stock diameter mm							
50	65	74	82	90	97	104	109	114
100	83	94	104	113	121	129	135	141
200	108	121	133	145	156	166	175	183
300	126	140	154	167	180	192	202	212
400	140	154	170	185	200	212	224	235
500	155	170	185	200	216	230	243	256
750	185	200	215	231	249	265	280	295
1,000	210	230	240	255	275	294	311	326
2,000	—	295	305	326	352	376	398	418
2,500	—	318	330	352	380	407	432	453
3,000	—	340	355	377	407	434	459	485
3,500	—	360	375	396	428	457	484	510
4,000	—	380	395	414	449	479	507	535
4,500	—	395	414	430	466	499	528	558
5,000	—	406	428	446	484	517	548	578
5,500	—	420	442	462	500	535	567	598
6,000	—	434	456	476	516	553	585	610

Per cent increase in rudder stock diameter for ships navigating in ice conditions

Class Л	10	10	10	8	6	4	2	0
Class УЛ	25	25	25	20	15	10	5	0

Note: The rudder stock diameter of oil tankers and tugs is to be increased by 10 per cent over the tabular values.

After determining the diameter of the rudder stock, the size of chain required and the tiller length (or quadrant radius) can be found in Table 52.

Table 52

Rudder stock diameter, d_{rs}, mm	Tiller length or radius of quadrant, R_0, mm	Size of open-link tiller chain, d_0, mm	Size of tiller rod-and-chain, d_0, mm
50	750	6	8
75	900	11	14
100	1,000	15	19
125	1,100	19	23
150	1,200	22	27
175	1,300	25	30
200	1,450	29	35
225	1,600	31	39
250	1,800	34	43
275	2,000	37	46
300	2,250	40	50

Note: If the radius of the quadrant or length of the tiller does not conform to the tabular values, the size of the chain is determined from the formula $d_c = d_0 \sqrt{\frac{R_0}{R}}$ in which d_0 = the tabular size corresponds to the actual rudder stock diameter, R_0 = tabular radius of the quadrant and R = actual radius of the quadrant. The size of a tiller rod-and-chain is selected from the table according to the obtained value of d_c.

The diameter, d_{gp}, of the chain guide pulleys is taken equal to from 12 to 16 chain bar diameters, d_c, for an open-link chain cable and from 14 to 20 chain bar diameters, d_c, for a rod-and-chain cable.

The loss in the tiller chain guide pulleys depends upon the angle of contact of the chain on the pulley and is evaluated by the pulley efficiency, η_{gp}. Values of the latter are listed in Table 53.

Table 53

Pulley diameter, d_{gp}	Value of η_{gp} for an angle of contact of		
	45°	90°	180°
20 d_c	0.95	0.92	0.88
15 d_c	0.93	0.90	0.85

The radius of a tiller drum should be

$$R_{cd} = (8 \text{ to } 10)\, d_c \tag{336}$$

where d_c = diameter of the chain bar.

If a pocketed sheave or chain sprocket (Fig. 159) is used instead of a drum, the sheave radius, R_{ps}, for use with a calibrated (pitched)

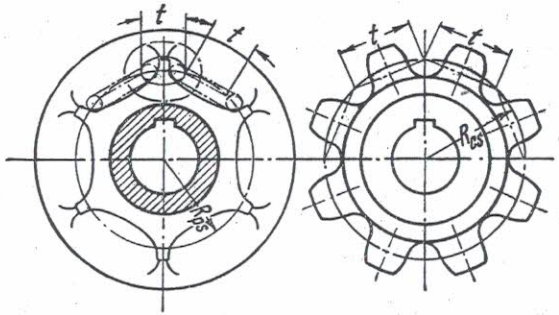


Fig. 159.

chain, or the sprocket radius, R_{cs}, for use with roller chain is

$$R_{ps} = \frac{1}{2}\sqrt{\left(\frac{t}{\sin\frac{90^\circ}{z_s}}\right)^2 + \left(\frac{d_c}{\cos\frac{90^\circ}{z_s}}\right)^2} \tag{337}$$

and

$$R_{cs} = \frac{1.75\sqrt[3]{T_s}}{\sin\frac{180^\circ}{z_s}} \quad \text{cm} \tag{338}$$

where $t = 2.6\, d_c$ = pitch of link chain, m

$z_s = 7$ to 9 = number of pockets or teeth

T_s = tangential force acting on the sprocket, kg.

The rpm, n_{cd}, of the tiller drum or n_s of the pocketed sheave or chain sprocket is found from the condition that the travel of the tiller is equal to the length of tiller chain being heaved in. Thus:

(a) for a drum we obtain

$$\frac{2\pi R_0 2\alpha^\circ}{360\tau} = \frac{2\pi R_{cd} n_{cd}}{60} \quad \text{whence } n_{cd} = \frac{R_0}{R_{cd}}\frac{\alpha^\circ}{3\tau^\circ} \tag{339}$$

(b) for a pocketed sheave or sprocket

$$\frac{2\pi R_0 2\alpha^\circ}{360\tau} = \frac{atn_s}{60} \quad \text{whence } n_s = \frac{2\pi R_0}{3at}\frac{\alpha^\circ}{\tau} \tag{340}$$

where a = number of links engaged in one revolution (for a calibrated link chain $a = 2z_s$, while for a roller chain $a = z_s$)

t = pitch of link or roller chain.

As a first approximation, the radius of the worm wheel can be determined from the formula:

$$R_{ww} = 0.294\sqrt[3]{\frac{M_{ww} z^2_{ww}}{A_t}} \quad \text{cm} \tag{341}$$

where M_{ww} = torque acting on the worm wheel shaft, kg-cm

z_{ww} = number of teeth on the worm wheel, equal to at least 30 (usually 40 to 100) and depending upon the speed of the motive unit shaft

A_t = a factor depending upon ψ_t, which is the ratio of the length, b, of the tooth to the circular pitch, t, and upon the assigned allowable unit bending stress, R_b, of the material of which the wheel is made.

For the gearing employed in steering engines $\psi_t = \frac{b}{t}$ ranges from 2 to 2.5, and factor A_t (Table 54) is calculated from the following formula, depending upon the material of which the teeth are made:

$$A_t = \frac{R_b}{15}\frac{b}{t} = c_t\psi_t \quad \text{kg per sq cm}$$

where R_b = allowable unit bending stress, kg per sq cm

$c_t = \frac{R_b}{15}$

Table 54

Worm wheel teeth material	A_t, kg/cm²	R_b, kg/cm²
Cast iron	40 to 50	300 to 370
Bronze	85 to 100	640 to 700
Cast steel	95 to 130	700 to 800
Forged steel	100 to 150	750 to 900

If the gearing ratio i_{wg} of the worm gearing is known, the radius of the pitch circle of the worm is

$$\rho = \frac{R_{ww}}{i_{wg} \tan \beta_p^\circ} \tag{342}$$

The required circular pitch, t_{pg}, of a pair of mating spur gears can be determined from the tangential force, T_{pg}, acting at the pitch diameters of their teeth:

$$t_{pg} = \sqrt{\frac{T_{pg}}{A_t}} \text{ cm} \tag{343}$$

The mean indicated power of the steam steering engine should be

$$N_{im} = \frac{N_m}{\eta_m} \tag{344}$$

where η_m = 0.8 to 0.85 = mechanical efficiency of a steam engine.

As a rule, the cylinder diameter, D_{cs}, of a steam engine for driving any kind of deck machinery, including the steering gear, is found from the following formula proposed by Academician V. Posdyunin. It is based on the conditions for starting the engine shaft from rest. Thus

$$D_{cs} = 2\sqrt[3]{\frac{M_m}{\pi\psi_r\eta_m(\alpha_i k_i p_{ts} - p_{ss})}} = 1.37\sqrt[3]{\frac{M_m}{\psi_r\eta_m(\alpha_i k_i p_{ts} - p_{ss})}} \text{ cm} \tag{345}$$

where M_m = torque developed on the engine shaft, kg-cm
α_i = 0.76 to 0.8 = indicator diagram factor
ψ_r = 0.80 to 1.3 = cylinder ratio
p_{ts} = design absolute pressure of the live steam admitted to the cylinder (taken from 3 to 4 atm lower than the boiler pressure)
p_{ss} = design absolute pressure of the spent steam, in atm abs
$k_i = \frac{1 + \ln \Delta}{\Delta}$ = coefficient of the mean theoretical indicated pressure for a ratio of expansion Δ.

Taking into consideration the loss in steam pressure in the pipelines, and starting and distribution valves and the steam leakage through the valves, as well as the necessity for ensuring dependable operation of the steering gear at low steam pressures, p_b, in the boiler and the fact that the selected value of α_i is only approximate, it is usually taken that

$$(k_i\alpha_i p_{ts} - p_{ss}) = (0.5 \text{ to } 0.8)\, p_b \tag{346}$$

If the cylinder diameter obtained in formula (345) is less than 80 mm, thereby leading to difficulties in engine manufacture, the pressure of the steam admitted to the steering engine is lowered by means of reducing valves. In this case, denoting the reduced steam pressure as p_r,

$$(k_i\alpha_i p_{ts} - p_{ss}) = (0.5 \text{ to } 0.7)\, p_r \tag{347}$$

Since the engine operates without steam expansion there are no particular advantages in the use of high steam pressures, from the point of view of operational efficiency. On the other hand, the use of lower steam pressures reduces steam leakage, reduces the stresses to which the parts of the steering engine are subjected and, consequently, their size, simplifies the design of the packing and increases operational safety. The torque developed on the shaft of the steering engine can easily be increased by adjustment of the reducing valve to increase the steam pressure if the latter proves insufficient in actual operation.

The pressure of the spent steam is taken to be:
p_{ss} = 1.2 atm abs if the spent steam is exhausted into the atmosphere,
p_{ss} = 0.2 to 0.3 atm abs if exhaust is into a condenser,
p_{ss} = 1.2 to 3 atm abs if the spent steam is used to heat the boiler feed water.

The indicator diagram factor, α_i, depends upon the mean velocity, c_m, m per sec of the piston, as is evident in Table 55.

Table 55

c_m	0.25	0.5	0.75	1.00	1.25	1.50	1.75	2.00	2.25	2.5	2.75	3.00
α_i	0.88	0.79	0.70	0.62	0.58	0.54	0.48	0.45	0.42	0.4	0.39	0.38

The indicated power required to start the steering engine shaft will be

$$N_i = \frac{\pi\psi_r(\alpha_i k_i p_{ts} - p_{ss})}{450}\left(\frac{D_{cs}}{10}\right)^3 n \text{ metric hp} \tag{348}$$

The reserve power is

$$\varphi_{res} = \frac{N_i}{N_{im}} \approx 1.27$$

Formula (345) can be transposed so that it may be used to determine the torque developed on the rudder stock by the steering engine. Steering gear data is measured and substituted into the formula.

It should be noted that in Posdyunin's formula (345) the indicator diagram factor, α_i, and k_i could be more correctly referred, not to p_{ts}, but to $p_{ts} - p_{ss}$. Then, combining with equation (312), we obtain

$$D_{cs} = 2\sqrt[3]{\frac{M_m}{\pi\psi_r\alpha_i k_i (p_{ts}-p_{ss})\eta_m}} = 2\sqrt[3]{\frac{M_{rs}}{\pi\psi_r\alpha_i k_i (p_{ts}-p_{ss})\eta_m\eta_{sg}i_{sg}}} \quad (350)$$

then

$$M_{rs} = \frac{D^3_{cs}\pi\psi_r\eta_m\eta_{sg}i_{sg}\alpha_i k_i (p_{ts}-p_{ss})}{8} \text{ kg-cm} \quad (351)$$

After measuring the stroke, S, of the steering engine piston and the speed, n_m, of its shaft, we can determine the mean velocity of the piston:

$$c_m = \frac{Sn_m}{30}$$

Using this value we can find the indicator diagram factor in Table 55.

The pressures of the live and spent steam are measured by gauges while an approximate value of the mechanical efficiency, η_m, is taken within the range established for steering gears.

Next, the values determining the gearing ratios of the elements in the gear train of the steering gear are measured, mean efficiency values are taken for these elements and then the total gearing ratio of the steering gear is calculated, together with its overall efficiency.

As a result of this procedure, we can determine the torque developed on the rudder stock by the steering engine, using formula (351).

The approximate steam consumption of the steam engine will be

$$G_s = \alpha_u g_i N_i \text{ kg per hour} \quad (352)$$

where α_u = 0.25 to 0.3 = utilization factor of the steering engine in regard to time and power during normal running conditions

N_i = indicated power of the engine

g_i = specific steam consumption, kg per ihp-hr, equal to:
25 to 30 for exhaust into feed-water heaters
25 to 22 for exhaust into atmosphere
22 to 20 for exhaust into a condenser.

In determining the required power of electric motors for driving steering engines, it is necessary to take into consideration the fact that they can operate for a certain period with an overload and that the speed varies with the load. Therefore, the power calculated on the basis of the mean speed of the motor shaft will be too high.

This ability of the motor to vary its speed with the load is taken into account by the factor a_e which ranges from 1.15 to 1.35. Then the gearing ratio of the steering gear is

$$i_{sg} = \frac{n_r}{a_e n_{rs}} = \frac{n_r}{a_e \frac{\alpha^\circ}{3\tau}} = \frac{3\tau}{\alpha^\circ}\frac{n_r}{a_e} \quad (353)$$

where n_r = rated speed of the electric motor as listed in the catalogue.

The required torque on the electric motor shaft is

$$M_r = \frac{M_{rs}b_e}{i_{sg}\eta_{sg}} \text{ kg-m} \quad (354)$$

where b_e = 0.5 to 0.7 = a factor taking into account the ability of the motor to operate with an overload.

The rated power developed by the motor, at a torque M_r expressed in kg-cm, is

$$N_r = \frac{M_r n_r}{716.20} \text{ metric hp} \quad (355)$$

The following calculation sequences may be employed in the design of steering gears with various types of engines and transmissions.

4-5. Power Calculations for Steam Steering Engines

Power calculations of a steering gear are usually carried out to determine the forces and torques acting on the separate elements of the gear, these values being used to determine the dimensions of the elements. It is necessary, in addition, to determine the required power, main dimensions and speed of the motive unit which ensure that the rudder is put over in the specified time.

Power calculations for a tiller-quadrant type of gear with a tiller chain transmission (Fig. 160) may be carried out in the following order.

1. The design torque, M_{rs}, acting on the rudder head is determined from the formula:

$$M_{rs} = \varepsilon M_0$$

where M_0 = torque acting on the rudder stock, kg-cm

ε = a factor taking into account the friction loss in the rudder hinges (ε = 1.05 to 1.10 for bottom-supported rudders and ε = 1.15 to 1.2 for spade rudders).

2. The tension in the tiller chain is

$$T_1 = \frac{M_{rs}}{R_0} \text{ kg}$$

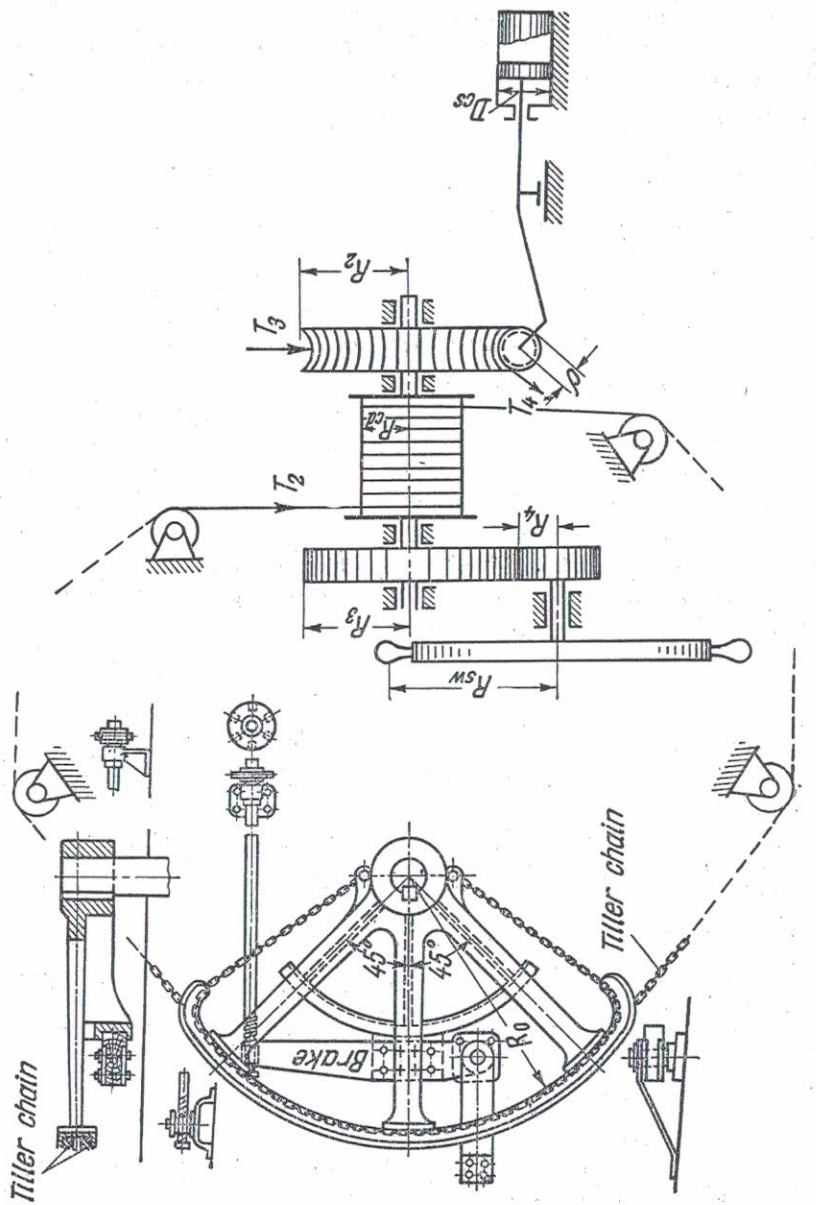


Fig. 160.

3. If the number of pulleys that the tiller chain runs over is denoted as k, the tangential force acting on the chain sprocket will be

$$T_2 = \frac{T_1}{\eta_{gp}^k} = \frac{M_{rs}}{R_0 \eta_{gp}^k} \quad \text{kg}$$

4. The radius of the tiller drum, R_{cd}, chain sprocket R_{cs} (for roller chain) or pocketed sheave R_{ps} (for link chain) is found from one of the following formulas:

$$R_{cd} = (8 \text{ to } 10)\, d_c \text{ cm or } R_{cs} = \frac{1.75 \sqrt[3]{T_2}}{\sin \frac{180^\circ}{z_s}} \text{ cm or}$$

$$R_{ps} = \frac{1}{2} \sqrt{\left(\frac{l}{\sin \frac{90^\circ}{z_s}}\right)^2 + \left(\frac{d_c}{\cos \frac{90^\circ}{z_s}}\right)^2} \text{ cm}$$

5. The torque acting on the drum, chain sprocket or pocketed sheave is

$$M_1 = R_{cd} T_2 \text{ or } M_1 = R_{cs} T_2 \text{ or } M_1 = R_{ps} T_2 \quad \text{kg-cm}$$

6. The torque acting on the shaft of the drum, chain sprocket or pocketed sheave is

$$M_2 = \frac{M_1}{\eta_{cd}} \text{ or } M_2 = \frac{M_1}{\eta_{cs}} \text{ or } M_2 = \frac{M_1}{\eta_{ps}} \quad \text{kg-cm}$$

7. The radius of the worm wheel is

$$R_2 = 0.294 \sqrt[3]{\frac{M_2 z_2^2}{A_t}} \quad \text{cm}$$

8. The tangential force acting on the teeth of the worm wheel at the pitch diameter is

$$T_3 = \frac{M_2}{R_2} \quad \text{kg}$$

9. Taking into consideration the friction loss in the bearings, the efficiency of the worm gearing is

$$\eta_{wg} = \eta_{wb} \frac{\tan \beta_p^\circ}{\tan (\beta_p^\circ + \varphi_f^\circ)}$$

10. The tangential force acting on the threads of the worm at the pitch circle is

$$T_4 = T_3 \tan (\beta_p^\circ + \varphi_f^\circ) \quad \text{kg}$$

11. The gearing ratio of the worm gearing is

$$i_{wg} = \frac{z_2}{z_w}$$

where z_w = number of threads on the worm.

12. The radius of the pitch circle of the worm is

$$\rho = \frac{R_2}{i_{wg} \tan \beta_p^\circ} \text{ cm}$$

13. The overall efficiency of the steering gear is

$$\eta_{sg} = \eta_{qt} \eta_{gp}^k \eta_{cd} \eta_{wg} \text{ or}$$
$$\eta_{sg} = \eta_{qt} \eta_{gp}^k \eta_{cs} \eta_{wg} \text{ or}$$
$$\eta_{sg} = \eta_{qt} \eta_{gp}^k \eta_{ps} \eta_{wg}$$

14. The torque acting on the worm shaft of the steering engine is

$$M_m = \rho T_4 \text{ kg-cm}$$

15. The speed of the chain drum, n_{cd}, chain sprocket, n_{cs}, or pocketed sheave, n_{ps}, is found from one of the following formulas:

$$n_{cd} = \frac{R_0}{R_{cd}} \frac{\alpha^\circ}{3\tau} \text{ rpm or}$$

$$n_{cs} = \frac{2\pi R_{cs}}{at} \frac{\alpha^\circ}{3\tau} \text{ rpm or}$$

$$n_{ps} = \frac{2\pi R_{ps}}{at} \frac{\alpha^\circ}{3\tau} \text{ rpm}$$

16. The speed at which the tiller chain runs over the drum, sprocket or pocketed sheave is

$$v_{tc} = \frac{2\pi R_{cd} n_{cd}}{60} \text{ or } v_{tc} = \frac{at n_{cs}}{60} \text{ or}$$

$$v_{tc} = \frac{at n_{ps}}{60} \text{ m per sec}$$

17. The speed of the worm (engine shaft) is

$$n_m = i_{wg} n_{cd} \text{ or } n_m = i_{wg} n_{cs} \text{ or}$$
$$n_m = i_{wg} n_{ps} \text{ rpm}$$

18. The overall gearing ratio of the steering gear is

$$i_{sg} = 3 n_m \frac{\tau}{\alpha^\circ}$$

19. The gearing ratio of the steering engine is

$$i_{se} = \frac{n_m}{n_{cd}} \text{ or } i_{se} = \frac{n_m}{n_{cs}} \text{ or } i_{se} = \frac{n_m}{n_{ps}}$$

20. The mean power applied to the shaft of the steering engine is

$$N_e = \frac{M_m n_m}{71{,}620} \text{ metric hp}$$

21. The mean required indicated power is

$$N_{im} = \frac{N_e}{\eta_m} \text{ metric hp}$$

22. The steam cylinder diameter of the steering engine is

$$D_{cs} = 2 \sqrt[3]{\frac{M_m}{\pi \psi_r \eta_m (\alpha_i k_i p_{ts} - p_{ss})}} \text{ cm}$$

23. The indicated power required to start the engine from rest is

$$N_i = \frac{\psi_r \pi (\alpha_i k_i p_{ts} - p_{ss})}{450} \left(\frac{D_{cs}}{10}\right)^3 n_m \text{ metric hp}$$

24. The reserve power is

$$\varphi_{res} = \frac{N_i}{N_{im}}$$

25. The speed of the steering wheel for hand steering is

$$n_{sw} = \frac{60 v_{sw}}{\pi D_{sw}} \text{ rpm}$$

26. The overall gearing ratio for hand steering is

$$i_{hs} = 3 n_{sw} \frac{\tau}{\alpha_{hs}^\circ}$$

27. The efficiency of the hand steering gear is

$$\eta_{hs} = \eta_{sw} \eta_{gp}^k \eta_{pg}$$

28. The torque acting on the steering wheel shaft is

$$M_{sw} = \frac{M_{rs}}{i_{hs} \eta_{hs}} \text{ kg-cm}$$

29. The force applied to the steering wheel rungs is

$$T_{sw} = \frac{2 M_{sw}}{D_{sw}} \text{ kg}$$

30. The force that must be exerted by each helmsman is

$$p_{hm} = \frac{T_{sw}}{k_{hm}} \quad \text{kg}$$

The power calculations for a quadrant drive with toothed gearing (Fig. 161) differs from the preceding only as to the first five items. These items follow.

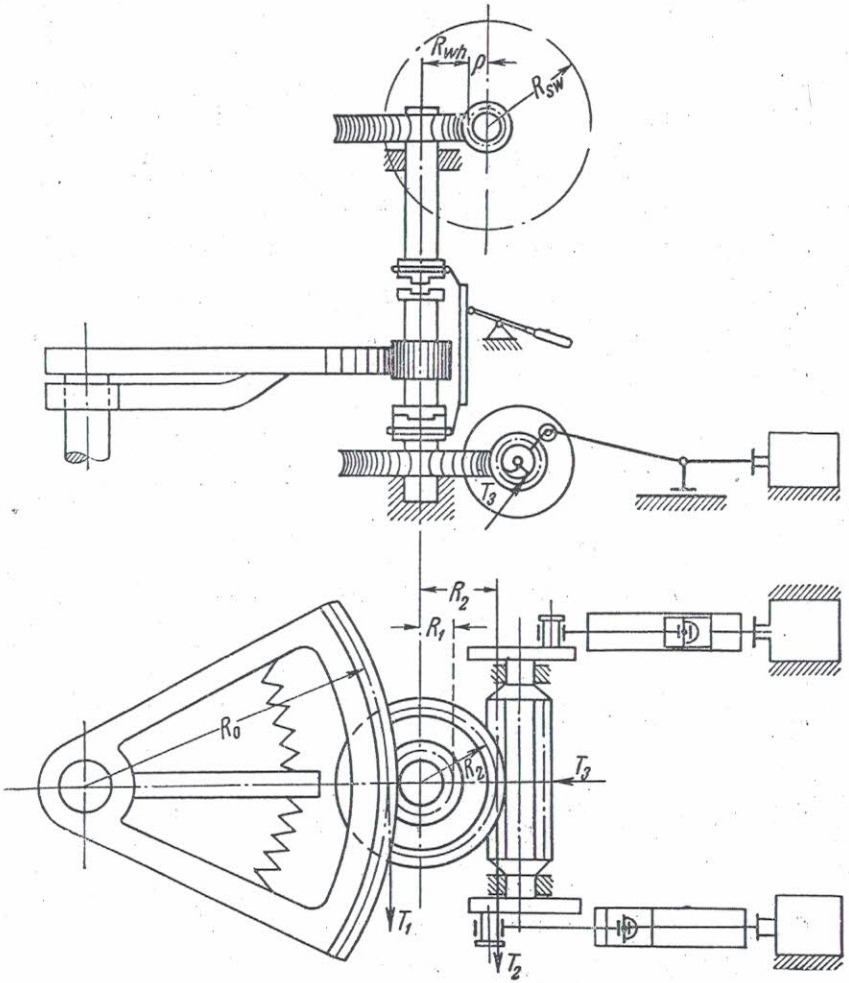


Fig. 161.

1. The tangential force acting on the main pinion at its pitch diameter is

$$T_1 = \frac{M_{rs}}{R_0 \eta_{qp}} \quad \text{kg}$$

2. The circular pitch of the quadrant and its spur pinion is

$$t_1 = \sqrt{\frac{T_1}{A_t}} \quad \text{cm}$$

3. The module of this gearing is

$$m = \frac{t_1}{\pi} \quad \text{cm}$$

4. If the quadrant has a central angle of $2\alpha°$, the number of teeth on its periphery is

$$z_q = \frac{2\pi R_0}{t_1}\frac{2\alpha°}{360°} = \frac{2\pi R_0}{\pi m}\frac{2\alpha°}{360°} = \frac{R_0}{m}\frac{\alpha°}{90°}$$

5. The pitch radius of the main pinion is

$$R_1 = \frac{z_1 t_1}{2\pi} \quad \text{cm}$$

where z_1 = 15 to 20 = number of teeth on the main pinion.

In this case, hand steering is done through worm gearing and may be calculated in the following order.

1. The torque, M_{wh}, developed on the worm wheel axis of the hand steering gear is

$$M_{wh} = \frac{R_1 T_1}{\eta_{qp}\eta_{ws}} \quad \text{kg-cm}$$

2. The pitch radius of the worm wheel is

$$R_{wh} = 0.294 \sqrt[3]{\frac{M_{wh} z_{wh}^2}{A_t}} \quad \text{cm}$$

3. The gearing ratio of the worm gearing is

$$i_{wh} = \frac{z_{wh}}{z_w}$$

where z_w = number of threads on the worm.

4. The pitch radius of the worm is

$$\rho = \frac{R_{wh}}{i_{wh} \tan \beta_p^\circ} \quad \text{cm}$$

5. The efficiency of the worm gearing is

$$\eta_{wg} = \eta_{wb} \frac{\tan \beta_p^\circ}{\tan (\beta_p^\circ + \varphi_f^\circ)}$$

6. The efficiency of the steering gear (for $\eta_{qp} \approx \eta_{pg}$) is

$$\eta_{hs} = \eta_{qp} \eta_{pg}^c \eta_{vs} \eta_{wg}$$

where c = number of pairs of spur gears in the gearing.

7. The torque on the worm or steering wheel shaft is

$$M_{sw} = \frac{M_1}{i_{hs} \eta_{hs}} \text{ kg-cm}$$

8. The force that must be exerted by each helmsman is

$$p_{hm} = \frac{M_{sw}}{R_{sw} k_{hm}} \text{ kg}$$

4-6. Determining the Main Data on Hydraulic Steering Engines

One of the principal problems to be solved in designing a hydraulic steering engine is to determine the diameter of the ram cylinders.

Certain investigators in this field have proposed methods involving consecutive trial-and-error computations for this purpose. Procedures of this nature, however, are not very convenient in practice. The method to be described attacks this problem in a more rational manner and aims to provide a more convenient procedure for determining the principal dimensions of hydraulic steering gears. The formulas will be deduced for the hydraulic steering gear shown schematically in Fig. 162.

The pressure, p kg per sq cm, generated by the pump of the hydraulic steering gear, acts on the bottom of ram *5*. Under the action of force, T_1 kg, developed by pressure p, the ram moves together with the swivel crosshead *1* in which the shank *2* of the tiller is fitted with a slide fit. Thus, movement of the ram is transmitted to the tiller which puts over the rudder.

Let us suppose that the rudder has been put over to an angle of α° from the middle-line plane. In this case, the following forces are applied to the various parts of the steering gear:

Q = force exerted by shank *2* of the tiller on the swivel crosshead *1* due to pressure, P_r, of the water on the rudder, kg

N = supporting force of guide *3* due to the pressure of the shoe *4*, kg

$f_1 N$ = frictional force between the shoes *4* and guides *3*, kg ($f_1 \approx 0.08$ = coefficient of friction)

$f_2 Q$ = frictional force between the contacting surfaces of tiller shank *2* and crosshead *1*, kg ($f_2 \approx 0.1$ = coefficient of friction)

$\frac{4}{\pi} f_3 Q$ = frictional force between the contacting surfaces of the swivel trunnions and crosshead body, kg ($f_3 \approx 0.05$ = coefficient of friction)

T_{sb} = frictional force in the stuffing boxes of the cylinder packing, kg.

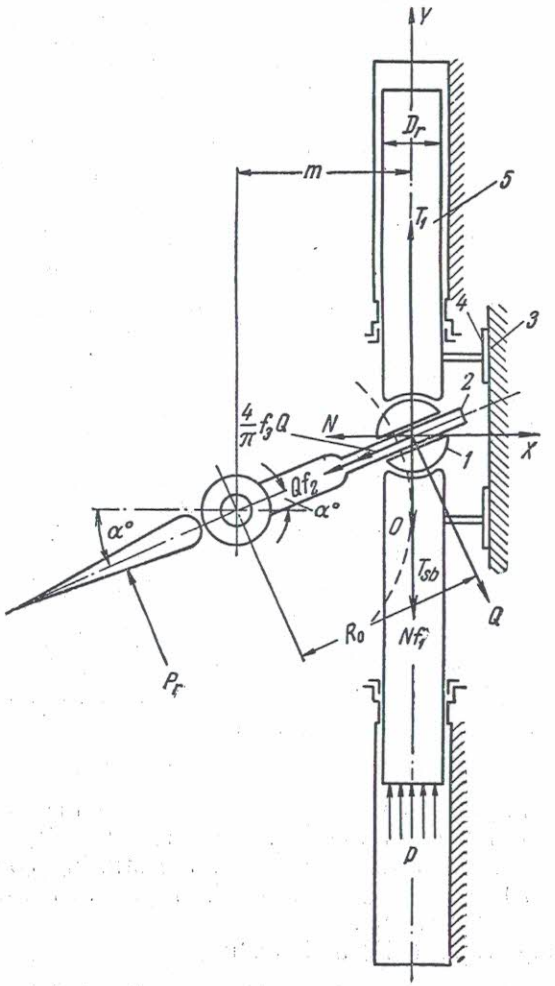


Fig. 162.

The force T_1 is the main design value on which the principal dimensions of the steering gear are based.

To determine the force T_1 we shall locate the origin of the coordinate system at the centre of the swivel crosshead. The plane system of forces shown in Fig. 159 is in equilibrium when the rams are in steady motion.

Projecting all the forces onto the vertical (y) axis we obtain

$$T_1 - T_{sb} - Q\cos\alpha^\circ - Nf_1 - Qf_2\sin\alpha^\circ - \frac{4}{\pi}f_3 Q\sin\alpha^\circ = 0 \quad (356)$$

After projecting all the forces on the horizontal (x) axis we obtain

$$-N + Q\sin\alpha^\circ - f_2 Q\cos\alpha^\circ - \frac{4}{\pi}f_3 Q\cos\alpha^\circ = 0 \quad (357)$$

After solving equation (357) for N and substituting its value into equation (356):

$$T_1 - T_{sb} - Q\cos\alpha^\circ - f_1\left(Q\sin\alpha^\circ - f_2 Q\cos\alpha^\circ - \frac{4}{\pi}Qf_3\cos\alpha^\circ\right) - \left(Qf_2\sin\alpha^\circ + \frac{4}{\pi}f_3 Q\sin\alpha^\circ\right) = 0$$

whence

$$T_1 = T_{sb} + Q\left[\left(1 - f_1 f_2 - \frac{4}{\pi}f_1 f_3\right)\cos\alpha^\circ + \left(f_1 + f_2 + \frac{4}{\pi}f_3\right)\sin\alpha^\circ\right]$$

Assigning mean values to the coefficients of friction from the previously indicated ranges and substituting we obtain

$$T_1 = T_{sb} + Q\left[(1 - 0.08\times 0.1 - 1.274\times 0.08\times 0.05)\cos\alpha^\circ + (0.1 + 0.08 + 1.274\times 0.05)\sin\alpha^\circ\right]$$

$$T_1 = T_{sb} + Q\,(0.987\cos\alpha^\circ + 0.244\sin\alpha^\circ) \quad (358)$$

The friction force, T_{sb}, in the stuffing boxes of the ram cylinders, for a coefficient of friction, f_{sb}, can be found from the following equation based on Mikheyev's formula:

$$T_{sb} = 0.15 z_r f_{sb}\pi D_r h_p p \text{ kg} \quad (359)$$

where z_r = number of rams in the steering engine
D_r = diameter of the ram plunger, cm
h_p = length of the packing in the stuffing box, cm
p = 70 to 150 = pressure of the liquid in the cylinders, kg per sq cm.

Combining equations (358) and (359):

$$T_1 = 0.15 z_r f_{sb}\pi D_r h_p p + Q\,(0.244\sin\alpha^\circ + 0.987\cos\alpha^\circ) \quad (360)$$

The force developed by the system of rams, expressed as a function of the pressure in the ram cylinders, is

$$T_1 = \frac{\pi D_r^2}{4} p z_n \quad (361)$$

where z_n = number of rams operating to develop the force T_1.

Combining equations (360) and (361) we can write

$$0.15 z_r f_{sb}\pi D_r h_p p + Q\,(0.244\sin\alpha^\circ + 0.987\cos\alpha^\circ) = \frac{\pi D_r^2}{4} p z_n$$

or

$$D_r^2 - 0.6 z_r f_{sb} h_p D_r \frac{1}{z_n} - \frac{4Q\,(0.244\sin\alpha^\circ + 0.987\cos\alpha^\circ)}{\pi p z_n} = 0 \quad (362)$$

Solving this quadratic equation for D_r, the ram plunger diameter, we obtain

$$D_r = \frac{0.6 z_r f_{sb} h_p}{2z_n} + \sqrt{\frac{(0.6 z_r f_{sb} h_p)^2}{4z_n^2} + \frac{4Q(0.244\sin\alpha^\circ + 0.987\cos\alpha^\circ)}{\pi p z_n}}$$

$$D_r = \frac{0.3 z_r f_{sb} h_p}{z_n} + \sqrt{\frac{(0.6 z_r f_{sb} h_p)^2}{4z_n^2} + \frac{16Q\,(0.244\sin\alpha^\circ + 0.987\cos\alpha^\circ)\,z_n}{4\pi p z_n z_n}}$$

$$D_r \frac{0.3 z_r f_{sb} h_p}{z_n} + \frac{0.3 z_r f_{sb} h_p}{z_n}\sqrt{1 + \frac{14.19Q\,(0.244\sin\alpha^\circ + 0.987\cos\alpha^\circ)\,z_n}{p\,(z_r f_{sb} h_p)^2}}$$

$$D_r = \frac{0.3 z_r f_{sb} h_p}{z_n}\left[1 + \sqrt{1 + \frac{14.19Q\,(0.244\sin\alpha^\circ + 0.987\cos\alpha^\circ)\,z_n}{p\,(z_r f_{sb} h_p)^2}}\right]$$

Assuming the maximum rudder angle $\alpha^\circ = 35^\circ$ for flat rudders and $\alpha^\circ = 32^\circ$ for hydrofoil rudders we obtain, respectively,

$$D_{35^\circ_r} = \frac{0.3 z_r f_{sb} h_p}{z_n}\left(1 + \sqrt{1 + \frac{13.45 Q z_n}{p\,(z_r f_{sb} h_p)^2}}\right) \text{ cm}$$

$$D_{32^\circ_r} = \frac{0.3 z_r f_{sb} h_p}{z_n}\left(1 + \sqrt{1 + \frac{13.7 Q z_n}{p\,(z_r f_{sb} h_p)^2}}\right) \text{ cm} \quad (363)$$

To determine the force exerted by the tiller shank on the swivel crosshead, we must first find the active length of the tiller

$$R_0 = \frac{m}{\cos\alpha^\circ} \quad (364)$$

where m = (2.5 to 3.2) d_{rs} = distance between the rudder stock and ram axes.

Then the force exerted by the tiller shaft on the swivel crosshead is

$$Q = \frac{M_{rs}}{z_n R_0} = \frac{M_{rs}}{m}\frac{\cos\alpha^\circ}{z_n} \quad (365)$$

Values of the coefficient of friction may be taken from Table 56.

Table 56

Surfaces	Coefficient of friction
Steel on steel in an oil bath	0.04
Steel on cast iron or steel, slightly lubricated	0.1
Steel on cast iron, dry	0.15 to 0.6
Steel on steel, dry	0.18 to 0.5
Bronze on cast iron, slightly lubricated	0.15
Bronze on steel, dry	0.18
Stuffing box with soft packing	0.06 to 0.11
Stuffing box of soft leather cups	0.03 to 0.07
Stuffing box of hard leather cups	0.11 to 0.13
Stuffing box of cups without lubrication	0.2

Assigning a tentative value to the thickness, δ_{sb}, of the packing, in cm, as measured radially in the ram cylinder, in the range from 30 to 60 mm, we can find the length of the packing from the empirical formula $h_p = (6 \text{ to } 8)\delta_{sb}$ cm.

After substituting the value of h_p into equation (363) we obtain the diameter of the ram cylinder.

Now we can check whether the assigned value of the packing thickness and its length are suitable for the calculated cylinder diameter, using the following formulas:

$$\delta_{sb} = (4.75 \text{ to } 7.9)\sqrt{D_r} \text{ cm}$$

$$h_p = (6.3 \text{ to } 28.5)\sqrt{D_r} \text{ cm}$$

The travel of each ram in putting the rudder from hard-over to hard-over is equal to the stroke of the ram:

$$S_r = 2R_0 \sin\alpha^\circ \text{ cm}$$

The mean speed of the ram during the time it takes to put the rudder from hard-over to hard-over is

$$c_r = \frac{S_r}{\tau} = \frac{2R_0 \sin\alpha^\circ}{\tau} \text{ cm per sec}$$

The volume of liquid, drawn in by the pump per second from z_n ram cylinders and delivered to the same number of cylinders in

putting over the rudder, is

$$Q_{sr} = z_n c_r \frac{\pi D_r^2}{4} = z_n \frac{2R_0 \sin\alpha^\circ}{\tau} \frac{\pi D_r^2}{4} = z_n \frac{m \sin\alpha^\circ}{\tau \cos\alpha^\circ} \frac{\pi D_r^2}{2} = z_n \frac{m\pi D_r^2}{2\tau} \tan\alpha^\circ$$

cu cm per sec (366)

If the steering gear is powered by a rotary radial-piston pump, the discharge per second of the pump will be

$$Q_{sp} = \frac{\eta_v}{60} z_p \frac{\pi D_p^2}{4} S_p n_p \text{ cu cm per sec} \quad (367)$$

where D_p = diameter of a pump cylinder, cm
z_p = 5 to 9 = number of pump cylinders
η_v = volumetric efficiency of the pump (Fig. 163)
S_p = stroke of the pump pistons, cm
n_p = 500 to 1,450 = rotational speed of the pump shaft, rpm.

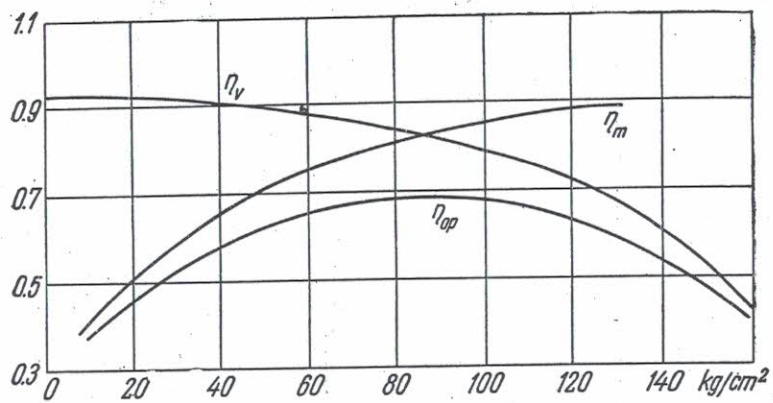


Fig. 163.

These pumps are designed with an odd number of cylinders so that the same number of cylinders draw in liquid from the rams as are delivering liquid to them and the volumes drawn in and delivered are the same (one cylinder is opposite the partition).

Since $\frac{S_p}{D_p} = \psi_p = 0.8$ to 1.6 and since $Q_{sp} = Q_{sr}$ we can convert equation (367) into the following:

$$Q_{sp} = \frac{\eta_v}{60} z_p \frac{\pi D_p^3}{4} \psi_p n_p \text{ cu cm per sec} \quad (368)$$

and

$$D_p = \sqrt[3]{\frac{240Q_{sp}}{\eta_v z_p \pi \psi_p n_p}} = 4.25 \sqrt[3]{\frac{Q_{sr}}{\eta_v z_p \psi_p n_p}} \text{ cm} \qquad (369)$$

The required power of the pump drive motor is

$$N_p = \frac{pQ_{sp}}{10^6} \gamma \frac{p10^4}{\gamma} \frac{1}{75\eta_{op}} = \frac{Q_{sp}p}{10^2 75\eta_{op}} \text{ metric hp} \qquad (370)$$

where p = 24 to 245 = pressure in the pump cylinders, kg per sq cm
γ = density of the liquid, kg per cu m
η_{op} = overall efficiency of the pump (Fig. 163).

Table 57 lists data on rotary radial-piston pumps that are being employed in hydraulic steering gears.

Table 57

Rotational speed, rpm	Maximum pressure, kg/cm²	Capacity, l/min	Power of drive motor, metric hp
523	140	416	—
580	155	340	—
625	175	246	—
675	175	197	—
750	175	148	—
750	70	500	75
800	70	250	37.5
850	70	120	18
850	175	125	—
850	175	83	—
850	70	63.5	9.5
900	70	33.5	—
900	210	43.5	—
900	230	21.8	—
1,000	70	12	2
1,100	210	28.4	—
1,160	210	28.4	—
1,160	245	14.2	—

Rotary pumps of other types (helical screw, vane, gear, etc.) used in hydraulic steering gears can be designed in accordance with the data presented in Part One of the present textbook. If a vane-type actuating gear is employed in the steering engine (Fig. 164), its principal dimensions can be determined by the method described below.

The diameter of the cylinder of the actuating drive in this case depends upon the torque that acts on the rudder stock and the pressure of the liquid on the vanes.

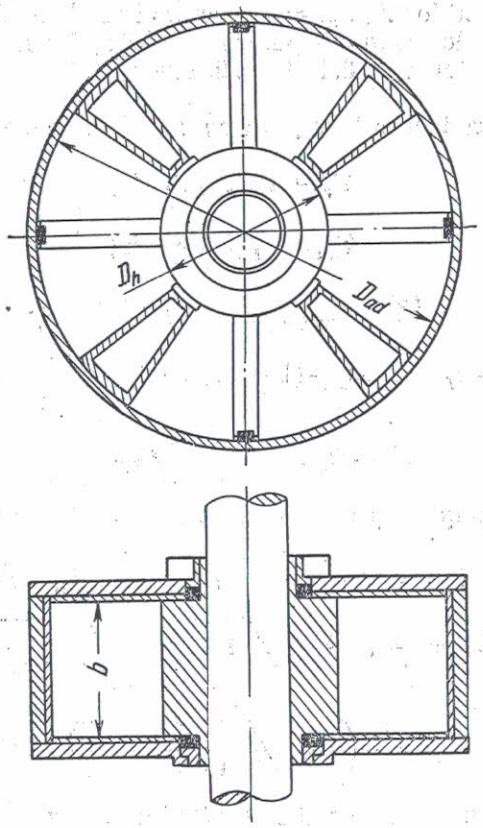


Fig. 164.

If a torque of M_{rs} kg-cm is applied to the rudder stock then the force on the vanes, due to this torque, will be

$$T_{rs} = \frac{4M_{rs}}{D_{ad}} \text{ kg} \qquad (371)$$

where D_{ad} = internal diameter of the actuating drive cylinder, cm.

Denoting the pressure in the cylinder as p kg per sq cm, the height of the cylinder as b (in m) and the number of vanes as z_v (z_v = 2 to 4),

the force developed by the liquid on the vanes is

$$T_1 = \frac{1}{2}(D_{ad} - D_h)\, bpz_v \tag{372}$$

where D_h = (1.6 to 2.2) d_{rs} = diameter of the hub of the vanes, cm.

The friction losses in the packing of the cylinder can be evaluated by the factor ε_f = 0.15 to 0.25 and therefore the friction force is $\varepsilon_f T_1$.

When the rudder is being put over and the vanes are in steady motion:

$$T_{rs} + \varepsilon_f T_1 = T_1 \text{ kg} \tag{373}$$

whence

$$T_{rs} = T_1(1 - \varepsilon_f) \text{ kg} \tag{374}$$

then

$$\frac{4M_{rs}}{D_{ad}} = \frac{1}{2}(D_{ad} - D_h)\, bpz_v (1 - \varepsilon_f) \tag{375}$$

$$(1 - \varepsilon_f)\, pbD_{ad}^2 z_v - (1 - \varepsilon_f)\, pbD_h D_{ad} z_v - 8M_{rs} = 0 \tag{376}$$

and finally

$$D_{ad}^2 - D_h D_{ad} - \frac{8M_{rs}}{(1 - \varepsilon_f)\, bpz_v} = 0 \tag{377}$$

Solving this quadratic equation, we obtain the cylinder diameter:

$$D_{ad} = \frac{D_h}{2} + \sqrt{\left(\frac{D_h}{2}\right)^2 + \frac{8M_{rs}}{(1 - \varepsilon_f)\, bpz_v}} \text{ cm} \tag{378}$$

$$D_{ad} = \frac{D_h}{2} + \frac{D_h}{2}\sqrt{1 + \frac{32M_{rs}}{(1 - \varepsilon_f)\, bpD_h^2 z_v}} \text{ cm} \tag{379}$$

thus

$$D_{ad} = \frac{D_h}{2}\left(1 + \sqrt{\frac{32M_{rs}}{(1 - \varepsilon_f)\, bpD_h^2 z_v}}\right) \text{ cm} \tag{380}$$

Next we can determine the required capacity of the pump that powers this steering engine. The velocity of rotation of the vanes is

$$\varphi_v^\circ = \frac{2\alpha^\circ}{\tau} \text{ deg per sec} \tag{381}$$

The volume of the cylinder swept through by the vanes per second is equal to the discharge of the pump. Thus

$$V_{sp} = \left(\frac{\pi D_{ad}^2}{4} - \frac{\pi D_h^2}{4}\right)\frac{1}{360}\varphi_v^\circ z_v b = \frac{\pi\varphi_v^\circ}{1{,}440}(D_{ad}^2 - D_h^2)\, z_v b =$$

$$= \frac{\varphi_v^\circ}{460}(D_{ad}^2 - D_h^2)\, z_v b \text{ cu cm per sec} \tag{382}$$

High pressures are not employed in the cylinder due to the difficulty in providing an effective packing for both cylinder and vanes. Ring or cup packing is commonly used for this purpose.

4-7. Operation of Steering Gears

Before starting a steam steering engine, it is necessary to open the corresponding valves in the live and exhaust steam lines, and then to blow through and gradually warm up the live steam line by slowly opening the blow-off and live steam valves.

Next, the rapid change-over from power to hand steering is to be checked. After inspecting to see that all foreign objects have been removed from the moving parts, the engine shaft is cranked over by hand to check again whether any foreign objects hinder shaft rotation or piston motion.

Then, all bolts, screws, nuts, locking devices and lubricating fittings are checked to see whether they are tight in place and whether the proper clearance has been provided in the hinged joints. The lubricating system is prepared for operation, lubricant is applied to all units, and parts requiring hand lubrication and the grease cups are filled.

After this, open the blow-off valves of the cylinders and the exhaust steam valve of the steering engine. The cylinders of the latter are blown through and warmed up by gradually opening the live steam valve. When the engine is sufficiently warm, turn its shaft one half revolution in each direction. Close the blow-off valves when condensate is no longer mixed with the steam issuing from the valves. To make sure that the steering engine is ready for action and operates normally, it must be started and run in each direction for several revolutions. Normal operation of the the whole steering gear and rudder indicator is checked by putting the rudder from hard-over to hard-over several times. At the same time, the action of the telemotors is to be checked, paying especial attention to the tightness of the pipeline joints, absence of backlash in hinged joints and coordination of the operation of the steering engine, telemotor and rudder by the readings of the rudder indicator and the rudder position.

If the ship is equipped with a hydraulic telemotor, the latter should be checked as described in Sec. 4-2 par. (A). Noise heard in the operation of a hydraulic telemotor indicates that air has penetrated into the system and that the actuating mechanism is not working properly. The system of a hydraulic transmission must be filled only with the liquid prescribed by the instructions. It should not be mixed with other oils, and cylinder and vegetable oils must never be used.

Items to be given attention in steam steering engine operation are: proper action of the starting and economic valves, normal lubricant supply, temperature of the parts subject to friction, absence of abnormally loud noises and absence of steam leakage. The action of the starting valve may be impaired by excessive backlash in the joints linking it to the control gear or by jamming in the valve itself. Incorrect operation of the starting valve leads to a lack of correspondence between the reading of the rudder indicator and the position of the rudder, or stops the steering engine.

Normal operation of the economic valve may be impaired by jamming of its moving parts due to cocking or excess wear. Other causes are incorrect adjustment of the spring tension, breakage of the spring or loss of its elasticity.

Special attention must be paid to timely closing of the economic valve and tightness between the lapped surfaces of the valve disk and its seat.

Abnormal operation of the economic valve will either stop the steering engine or increase the steam consumption.

A lack of sufficient lubricant supply to the friction surfaces may lead to overheating which may result in the melting of babbitted bearings and jamming of the drive mechanism.

Lubricating fittings should always be tightly closed except when lubricant is being applied; they should be cleaned periodically and the wicks of wick-type lubricators should be washed from time to time in a soda solution or kerosene.

Knocking in the drive mechanism of a steering engine shows that there is excess backlash in the joints linking the parts of the engine.

All defects discovered in the operation of a steering engine should be immediately eliminated if they may lead to serious trouble. If running conditions are such that it is impossible to stop the steering engine immediately for repairs, there being no danger of a breakdown so that the engine can continue to operate for some time, the engine should receive especial attendance until repairs can be made.

All trouble-shooting and repairs of the steering engine, before shutting off the steam supply, should be strictly prohibited.

During short stays of the ship in harbour it is necessary to close the live and exhaust steam valves and to open the blow-off valves of the steering engine. Additional procedure for prolonged stays in a port include the shutting of all valves in the live and exhaust steam lines, opening of all drain valves of the engine and steam line, shutting off lubricant supply to all points of lubrication and the removal of wicks from a wick-type system.

Only the liquid indicated in the operating instructions of the particular hydraulic steering engine should be employed as the working fluid. Mineral oils with a viscosity of 3 to 4.5 degrees Engler at 50°C, free of resin and acid, and retaining their properties at low ambient temperatures are usually employed. In many cases, however, oils suitable for northern latitudes cannot be used in southern seas and it is necessary to change the oil for navigating in different regions.

It is not permissible to mix various grades of mineral oil or to add animal or vegetable oils to obtain the desired properties. Water must be kept out of the system in charging the hydraulic steering engine by carefully filtering and settling the oil. The charging procedure must conform strictly to the instructions. The following general rules should be observed in charging the hydraulic engine with oil:

1. All cavities, chambers and passages of the steering engine that are to be filled with oil are connected together by opening the pertinent valves, as well as all nonreturn and stop valves and all air-relief valves and cocks.
2. The service tank or the steering wheel pillar (depending upon the type of engine) is charged with oil through the upper opening, and more oil is added as it runs down into the system.
3. Each successive air-relief valve is closed as the oil begins to flow from it.
4. The steering engine is started for a trial run with the air valves open, putting over the rudder 5° or 6° from side to side and using all pumps alternately, beginning with the hand pump.
5. The air-relief valves of the rams are closed as soon as liquid starts to flow from them.
6. During charging, the oil level in the service tank must not drop below the lower limit indicated on the sight gauge.

In addition to the usual measures consisting of inspection and checking the tightness of all joints and the soundness of various parts, a hydraulic steering engine is prepared for operation by checking the positions of all cocks, plugs and valves, and turning or shifting them to the positions prescribed by the instructions. All flange joints of the pipelines and fittings of the hydraulic system are to be tested for leakage. Operation of all the valves, especially the relief and bypass valves, is to be checked for jamming and other defects. The main and standby pumps are to be checked as well as the hand steering gear. All moving parts of the engine should be lubricated. The oil employed for this purpose should have a moderate viscosity; it should possess good lubricating properties, should not produce any residues, sludge, deposits or sediment, should be acid-free and separate readily from water, and it should not form stable emulsions. The engine guides are to be lubricated with high-viscosity oils (at least 8 degrees Engler). The distributing system to the points of lubrication is also prepared for operation.

Next, the engine is started for a trial run and the rudder is put from hard-over to hard-over consecutively from each control station, using each pump separately. During the test the working fluid system is checked for oil leakage and the operation of the parts of the steering gear is also checked.

When this preliminary procedure has been completed, the hydraulic steering engine is started in the following order:

(1) connect the main working pump to the steering gear system;

(2) switch on the controls of the steering engine pump from the control motor;

(3) switch on the control station from which the engine is to be controlled;

(4) start the drive motor of the main pump.

During steering engine operation check the lubricant supply to the friction surfaces, their temperatures, the pressure of the working fluid in the system, the oil level in the service tanks. Also check for leaks in the oil system and excess noise in operation of the engine.

To stop the steering engine, the rudder is first put over to the middle-line plane of the ship and then the engine is stopped. All the parts of the gear are carefully examined and the tightness of cotter pins and nuts is checked by tapping with a hammer. All necessary repairs are made. Dirt and used oil that have accumulated on the engine are removed and all friction and polished surfaces are coated with a thin layer of clean oil which is periodically renewed if the stay in port is prolonged.

Additional measures, required if the stay in port exceeds 30 days, are: coating all external polished and unpainted parts of the engine with a mixture of grease and white lead or chalk, and the friction surfaces with oil; stopping the supply of lubricant from all fittings and lubricating devices; and removing all instruments for storage in a dry, warm room.

The following steering engine troubles may be encountered:

1. The steering engine fails to put over the rudder as a result of:

(a) faults in the mechanical facilities of the control stations;

(b) faults in the lever system of the pump control mechanism;

(c) faults in the pump which fails to deliver liquid to the ram cylinders;

(d) lack of working fluid in the hydraulic system;

(e) the failure of liquid to circulate through the pipeline system due to faults in the relief-bypass valves of the rams.

2. The steering engine puts the rudder over only to one side and fails to return it to the central position due to faults in the relief-bypass valve in one branch of the pipeline system.

3. The rudder is put over very slowly because of a drop in the discharge of the pump.

4. The maximum angle to which the rudder is put over by the engine does not correspond with the hard-over position because the servomotor of the steering engine is out of adjustment.

5. The steering engine rams travel with nonuniform speed (with jerks) and noise is heard in the hydraulic system into which air has penetrated.

Air may get into the system through leaks in the pipe joints, rams, and packing due to a low level of liquid in the service tanks. Air is purged from the system through the air exhaust valves by pumping the working fluid through the system.

Chapter 5

ANCHORING AND WARPING MACHINERY

5-1. General Considerations

A ship in moorage or riding at anchor is subject to the following forces:

(1) force of the current on the bottom;
(2) force of the wind on the topside;
(3) inertia forces due to pitching and rolling.

Anchoring and warping, or mooring, arrangements serve to withstand these forces and to restrain the ship in a comparatively stationary position. The former are used when the ship is standing in a roadstead and the latter when tied to a wharf. Sometimes both are used at a berth.

The anchoring arrangement consists of anchors, anchor chain cable hawse holes, stoppers and anchor handling equipment.

In addition to the functions already mentioned, the warping appliances serve to haul the ship up to the wharf in mooring and consists of hawser ports, bollards, keep plates, fenders and warping mechanisms.

A ship is usually equipped with three types of anchors: bower, the main appliance used when riding at anchor; stream anchor, used for mooring the ship head and stern, and kedges used in getting a ship off a sand-bank.

A bower anchor is three times as heavy as a stream anchor which is twice as heavy as a kedge.

The anchoring arrangement must provide for:

(1) rapidly dropping the bower anchor, letting out the chain cable to the required depth and then smoothly stopping the chain;
(2) hoisting the anchor chain with the anchor;
(3) fixing the anchor chain reliably to the hull when riding at anchor and when under way, not allowing the chain to run out accidentally;
(4) readily stowing the anchor;
(5) rapidly releasing the anchor and chain from the hull and dropping them overboard;
(6) rapidly releasing the anchor from the chain.

The first two requirements are met by the anchoring machinery, the third by the stopper, the fourth by corresponding design of the anchor and other parts of the anchoring arrangements, the fifth by securing the chain to the hull with a chain slip and the sixth by securing the anchor to the chain through a joining-shackle.

The warping arrangement must reliably hold the ship at the wharf and bring in the warping hawser in mooring operations. The former of these requirements is met by securing the hawsers to the bollards and the latter by means of the warping machinery.

The following principal demands are made to the machinery of anchoring and warping arrangements:

(1) dependability and safety in operation;
(2) ability to start smoothly at full load;
(3) ability to maintain the proper torque when the speed, at which the anchor chain or warping hawser is being brought in, drops to a low value or even to zero;
(4) ability to hold the anchor suspended or the hawsers in any position in mooring in case of a failure of power supply to the motive unit;
(5) convenient controls, smooth running and the possibility of stepless speed adjustment of the motive unit shaft;
(6) relatively small weight and the least feasible amount of deck space occupied;
(7) economy in operation.

Machines for hoisting the anchor are called windlasses and anchor capstans. Machines for heaving in mooring hawsers and for warping the ship in mooring operations are called warping winches and warping capstans.

Frequently, the anchoring and warping machines are designed as a single unit performing both functions and satisfying the requirements made to them. In this case, they are called windlasses or anchoring and warping capstans.

According to their motive unit, anchoring and warping machines are classified as hand, steam, electric, hydraulic and internal combustion engine driven.

Hydraulic windlasses, capstans and warping winches are powered by fluid pressure which is generated by rotary nonreversible radial-piston pumps driven by constant-speed motive units, similar to the pumps used for hydraulic steering engines.

The regulations of the U.S.S.R. Shipping Register stipulate that anchoring and warping machinery with a mechanical drive must be equipped with a standby hand drive.

A general view of the anchoring arrangement of a transport ship with the anchor in running order (stowed) is shown in Fig. 165.

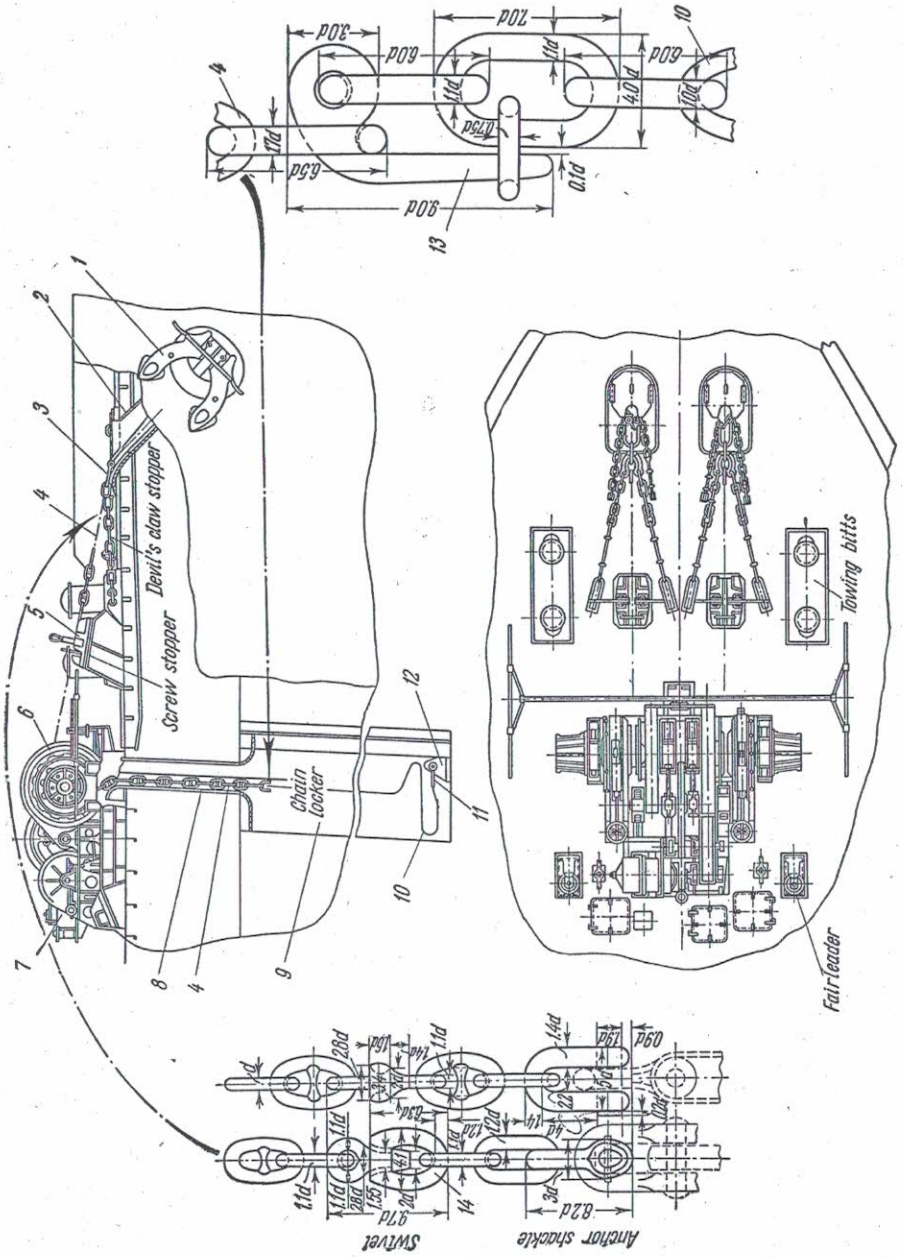


Fig. 165.

Anchor *1* has been pulled into the hawse hole *2* and is connected by joining-shackle *3* to the anchor chain cable *4*. The latter passes through stopper *5* which is fixed securely to the deck. After the stopper the chain runs over the cable lifter of windlass *6* which is driven by electric motor *7*. From the cable lifter the chain passes through the chain pipe *8* and is lowered into the chain locker *9* where it is secured to the chain slip *10*. The latter is connected by joining-shackle *11* to the cable clinch *12* which is securely fixed to the framing of the ship's hull.

A slip hook *13* joins the chain *4* to the chain slip *10* so that the anchor and chain can be quickly dropped overboard if necessary. Swivels *14* are provided at the first and last link to prevent the chain cable from twisting when the ship is riding at anchor.

An anchor chain is composed of separate 23 to 25 m lengths, called shots, joined together by shackles. The anchor chain consists of an uneven number of shots so that the joining-shackles bed in the same position in the whelps, or snugs, on the cable lifter. Each end of each shot has two enlarged links to ensure a smooth transition from the normal links to the shackle.

According to the motor speed and the torque required on the cable lifter, electric windlasses and capstans are designed to be driven either through one set of worm gearing, worm gearing and a pair of spur gears or worm gearing with two sets of spur gears. The trend at the present time is to use bevel and spur gearing of the ordinary and planetary type instead of worm gearing, due to the low efficiency of the latter.

Steam-driven windlasses are always designed with spur gearing while capstans may have both spur and worm gearing.

Windlasses may be designed with one or two intermediate shafts, depending upon the type and number of sets of gearing employed. The cable lifters are engaged to the windlass drive by shifting sliding gears, the lifters themselves, or clutches.

5-2. Construction of Anchoring and Warping Machinery

A steam-and-hand driven windlass with a double spur gear transmission is illustrated in Fig. 166. The windlass is mounted in frames *25* on the foundation plate *24* which rests on a wooden pad *15*. The windlass is powered by a two-cylinder duplex horizontal steam engine. It is controlled by starting valve *27* with lever *28*.

The starting valve operates in the same way as the corresponding valves of steam steering engines. All steam engines of anchoring and warping machinery operate without steam expansion.

The reciprocation of the pistons *29* in the cylinders *30* is transmitted by the piston rods *1* and connecting rods *2* to the crank disks

4 on the main shaft *5* of the steam engine. This shaft also carries eccentrics *3* which actuate the distribution valves *26*. Driving shaft *5* and shaft *8*, carrying the winch heads, or warp ends, *18* and *19*, are linked through gears *6* and *7* which are keyed to the shafts.

Claw-type sliding clutch members *10* are mounted on feathers on shaft *8* while gears *9* are freely mounted.

Jaws are provided on the end faces of both the sliding clutch members *10* and the gears *9*. The clutch members are engaged with or disengaged from gears *9* by the action of two fork-ended two-arm levers *20*. The lower fork of the levers has pins on which bronze slide blocks are mounted, these blocks entering annular slots on the clutch members *10*. Nuts *31* are mounted on trunnions in the upper forks of levers *20*. The threaded ends of rods *32*, on which handwheels *21* are mounted, pass through nuts *31*. Turning handwheel *21* moves nuts *31* along the thread of rods *32*, thereby turning levers *20* about pins *33* so that the clutch members *10* are shifted in the desired direction. Gear *9* meshes with gears *11* which slide along a feather on load (anchor) shaft *12*. One end face of the hubs of gears *11* has jaws which engage jaws on one end face of the hubs of cable lifters *13*. The latter are freely mounted on shaft *12*.

The lifters *13* are engaged with and disengaged from gears *11* and, consequently, shaft *12* by means of a device that is described in the following.

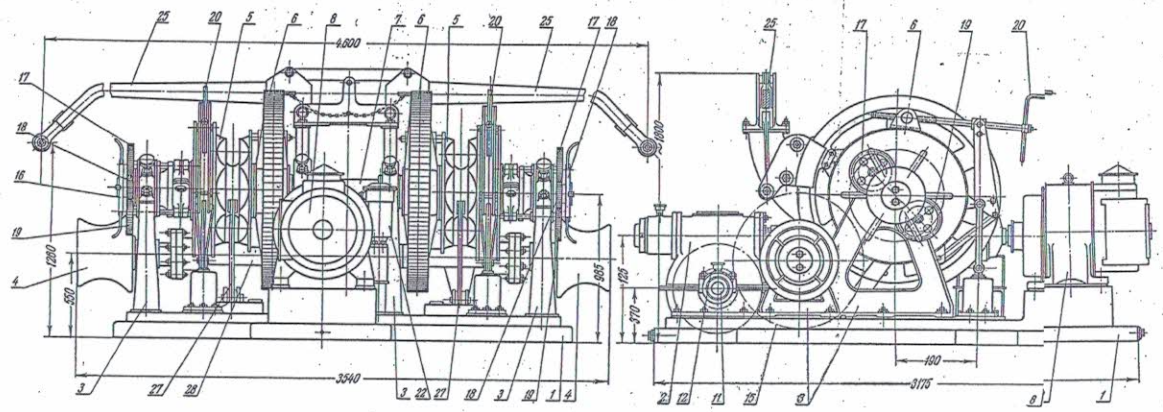
Threaded shafts *16*, having crank-handles *17*, pass through slide blocks *37*. Tie rods *60* link slide blocks *37* with angle levers *61*. The angle levers pivot about pin *62*. The other ends of the levers enter slots in the hubs of gears *11*.

When handles *17* are rotated together with shafts *16*, slide blocks *37* move along the shafts, actuating tie rods *60* and angle levers *61* which turn about pins *62*. This movement shifts gears *11* along shaft *12* in the desired direction.

The windlass has a hand-operated band brake in which the band *35* is lined with wooden blocks *36*. The brake band is applied in a groove of brake disks *34*. The latter are made integral with the cable lifters.

Nut *38*, hinged to the brake band, is secured on pin *39*. The threaded end of rod *40*, which is turned by crank handle *41*, passes through nut *38*. The middle, enlarged part of rod *40* passes through bushing *44* which is pivoted in shoe *42*. The latter turns about pin *43*. Pin *43* is fastened to the upper end of the brake band *35* and is linked through tie rod *46* to the foundation plate *24* of the windlass. The lower end of shoe *42* is linked through tie rod *45* with the lower end of the brake band.

Thus, when handle *41* of rod *40* is turned, the threaded end of the rod moves axially in nut *38* and swings shoe *42* about pin *43*. De-



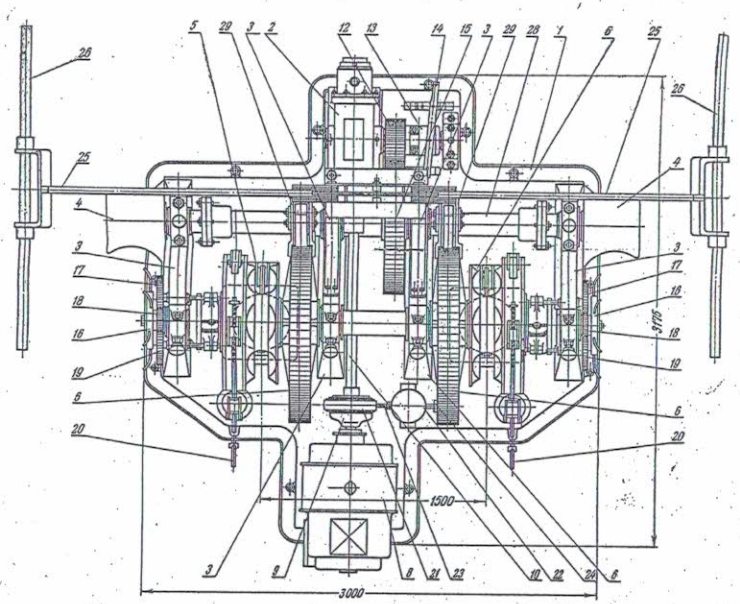
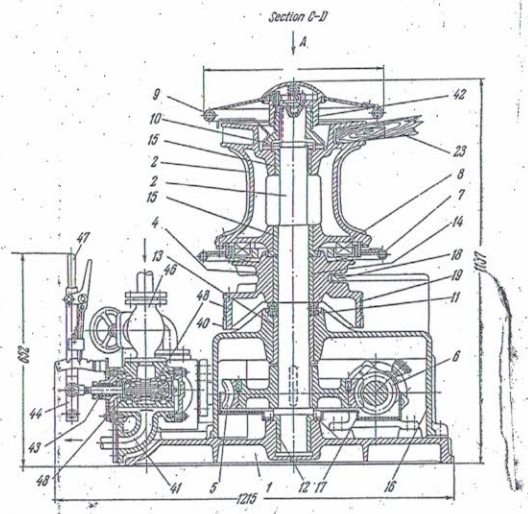


Fig. 167



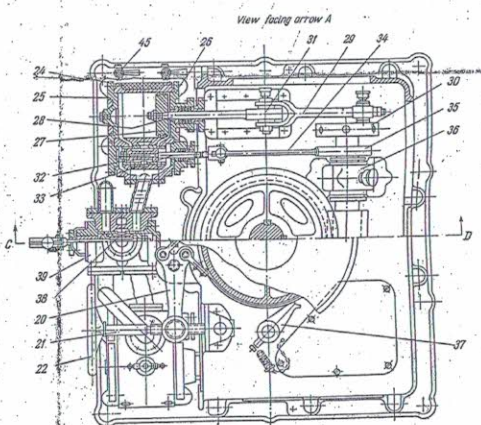


Fig. 168

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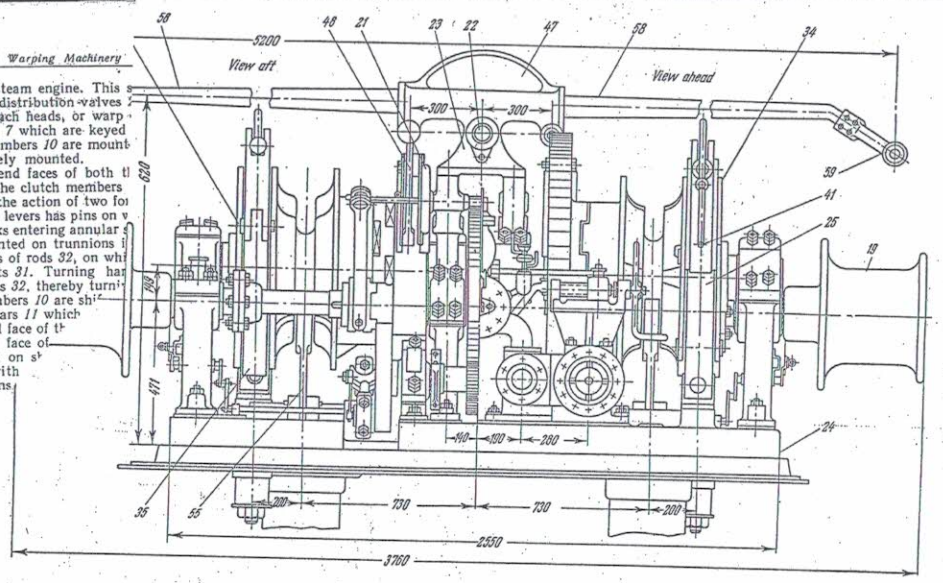
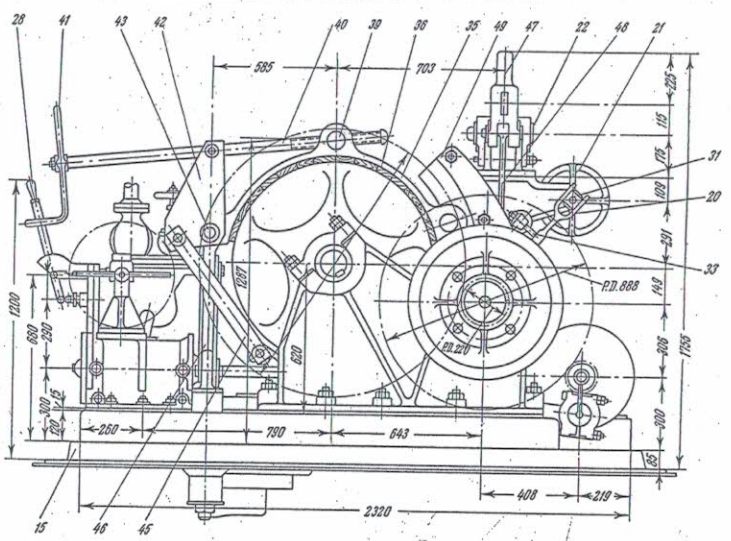
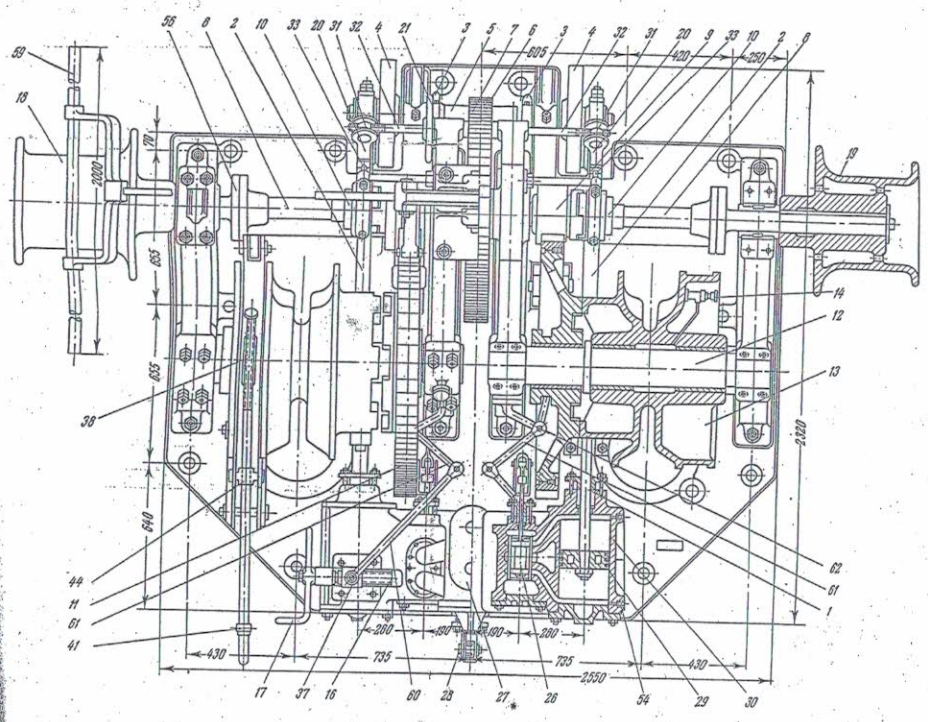






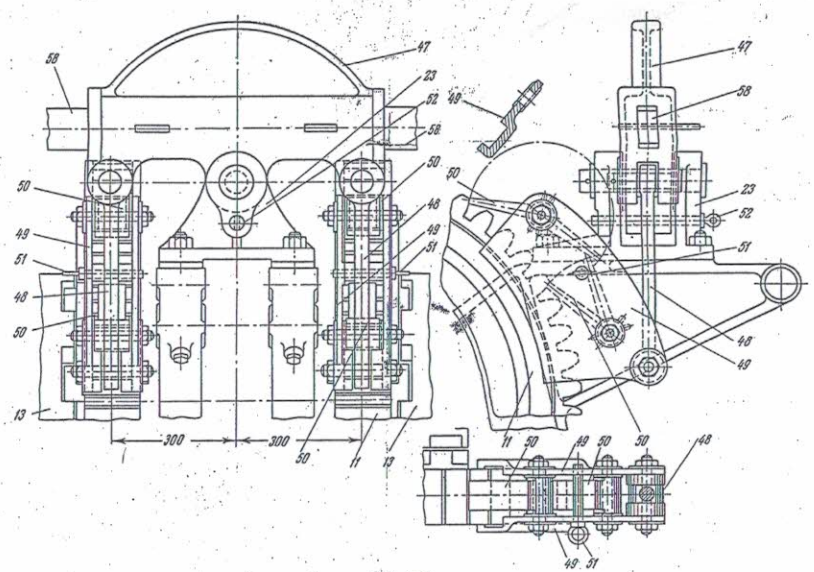


Fig. 166

pending upon the direction in which the handle is turned, this either applies or releases the brake.

The hand drive of the windlass is actuated by the rocker arm which consists of the centre-piece *47* into the sockets of which arms *58* are fitted.

The centre-piece *47* turns on pin *22* which rocks in bearings mounted in upright *23*. The upright is secured to the frames of the windlass. Handles *59* are fitted into arms *58*.

Centre-piece *47* is linked through two tie rods to two shoes having cheeks *49* between which pawls *50* are mounted on pins. The pawls engage the teeth of gears *11*. As the rocker arm is oscillated on pin *22*, one tie rod *48* moves downwards with its shoe while the other moves upwards.

The pawl of the shoe moving downwards slides over the teeth of its gear while the pawl moving upwards engages the teeth and turns its gear *11* and, with it, shaft *12*.

When the windlass is driven by the steam engine, the pawls are swung back and rest on cotter *51*, and the rocker arm is fixed in the horizontal position by cotter *52*.

To hoist one of the anchors the corresponding crank handle *17* is rotated until gear *11* is shifted into engagement with cable lifter *13*. Both gears are shifted into engagement with their cable lifters if both anchors are to be hoisted simultaneously. Clutch members *10* must also be shifted into engagement with gears *9*. Then, shaft *5* will drive shaft *12* through gears *6*, *7*, *9* and *11*. Warping shaft *8* will also rotate together with warp ends *18* and *19*. The warp ends are fastened to shaft *8* by flange joints to facilitate replacement in case of damage and also for convenience in assembly and manufacture of shaft *8*. If only warping operations are to be done, gears *11* are disengaged from cable lifters *13*. In this case, the warp ends will be driven from shaft *5* through gears *6* and *7*.

To hoist an anchor by hand, the corresponding gear *11* should be engaged to its cable lifter *13* while both clutch members *10* should be disengaged from gears *9*. Pawls *50* should be swung into engagement with the teeth of gears *11* and cotter *52* should be removed.

The valve chest of the starting valve is furnished with a relief valve to protect the steam engine against excess pressures in its cylinders. Threaded holes *54* are provided in the cylinders for mounting blow-off valves and drain cocks. The bearings of the cable lifters are lubricated through fittings *14*.

Throw-off pawls *55* prevent jamming of the chain links in the whelps of the lifter and extract jammed links.

An electric-and-hand driven windlass with one worm gearing and two spur gear transmissions is shown in Fig. 167. The parts and

units of the windlass are mounted in frames *3* on foundation plate *1*.

Through coupling *9* and shaft *10*, motor *8* drives the worm gearing which is enclosed in housing *2*. Coupling *9* carries an electromagnetic brake operated by solenoid *22*. Gear *12* is freely mounted on the worm wheel shaft while clutch member *13* can be shifted along this shaft on a feather by turning lever *14*. Jaws on end faces of gear *12* and clutch member *13* comprise a clutch to engage gear *12* with the shaft.

Gear *12* meshes with gear *15* which is keyed on shaft *28* of the warp ends *4*. Also keyed on shaft *28* are gears arranged under the shoes *29* of the hand drive and meshing with gears *6* keyed on the load shaft *7*. Thus, the load and warping shafts are driven by the motor when clutch member *13* is engaged with gear *12*.

Cable lifters *5* are freely mounted on load shaft *7*. To put them into operation they must be shifted along the shaft *7* so that the jaws on their end faces engage those on the hubs of gears *6*. The lifters are shifted by rotating handwheels *16* which operate through a transmission consisting of gears *17*, *18* and *19*.

The hand drive of the windlass is actuated by the handles *26* of a rocker arm *25*, and operates in the same manner as in the preceding windlass. To hoist the anchor by hand, clutch member *13* must be shifted to the position shown in Fig. 167, i. e., out of engagement with gear *12*.

The windlass has an ordinary band brake, applied by turning crank handle *20*, and an electromagnetic brake *21* operated by solenoid *22*. Lever *23* of the electromagnet brake has a weight *24*. The brake band is held in the released position as long as current flows in the supply circuit of the magnet. Upon power failure or if the current is switched off, the electromagnet is deenergized and the weight *24* applies the brake to the brake disk *9*. Throw-off pawls *27* have the same purpose as in the preceding windlass.

A steam-and-hand driven anchoring and warping capstan is illustrated in Fig. 168. It is powered by a duplex horizontal steam engine. The reciprocation of the engine pistons *27* is transmitted through piston rods *28*, crossheads *31* and connecting rods *29* to the crank disks *30* which are mounted on the shaft of worm *6* running in bearings *36*. Steam distribution is effected by a single-eccentric valve gear and piston valves *32*. The latter are actuated through valve rods *33*, tie rods *34* and straps *35* riding on eccentrics which are mounted on the shaft of worm *6*.

Live steam is admitted into the valve chest *39* of starting valve *38* through the steam valve *46* while spent steam is exhausted through connection *41*. Covers *25* and *26* close the ends of the cylinder which has insulation *24* to reduce thermal losses.

The starting valve is shifted by hand with lever *47* to the lower end *44* of which the rod *43* of the starting valve is linked.

In operation, the rotation of worm *6* is transmitted to its mating worm wheel *5* which is keyed on the capstan shaft *2*. The rotation of the capstan shaft, running in bronze bearings *12* and *13* fitted in base *1* and housing *16* of the worm gearing, is transmitted through the jaw clutch *10* to the capstan barrel *3*. The latter is freely mounted with its bushings *15* on capstan shaft *2*. The claw clutch *10* is engaged or disengaged by rotating handwheel *9*. This raises or lowers bushing *42* along a key in the capstan shaft and the thread in the handwheel hub. By means of jaw clutch *8*, barrel *3* can be engaged to cable lifter *4* into which bushing *14* is fitted. Cable lifter *4* is freely mounted on the shaft and rests on ball thrust bearing *11*. The worm gearing has a pan *17* for the oil bath. Sheave *48* is cast integral with cable lifter *4* and accommodates band *19* of the brake which is applied from handwheel *22* through threaded rod *21* and lever *20*. The lower part of the brake sheave *48* has a ratchet wheel *40* engaged by pawl *37* to prevent reverse rotation of the capstan barrel and cable lifter whenever necessary.

If clutch *10* is engaged and clutch *8* is disengaged, only drum *3* is driven by the steam engine. Both drum *3* and cable lifter *4* operate if clutch *8* is also engaged by turning handwheel *7*.

The capstan is changed over to the hand drive from the capstan bars *23* by disengaging clutch *10*. As before, only the barrel is operated if clutch *8* is disengaged while both barrel and cable lifters are operated if this clutch is engaged. The steam engine cylinders are equipped with blow-off valves *45*. Throw-off pawl *18* prevents piling of the chain cable in the hawse hole of the chain locker.

An electric-and-hand driven capstan with integral barrel and cable lifter is illustrated in Fig. 169. The capstan is mounted on base plate *1*. Capstan shaft *2* passes through bushings *10*, *11*, *12* and *18*. Its axial load is carried by ball thrust bearing *20*. Barrel *8* rests on thrust washer *9*.

Through flexible coupling *22* the motor *23* drives the worm shaft with worm *3* which meshes with worm wheel *4* keyed on shaft *5*. The latter runs in bearings *19* and *21*. Keyed on this shaft is gear *6* which transmits power to gear *7* keyed on the capstan shaft *2*. Rotation is transmitted from shaft *2* to the barrel *8* and cable lifter, mounted freely on the shaft, through head *13* and inserted pins *14*. Pawls *15*, freely mounted on pins *16* and secured by nuts *17*, prevent reverse rotation of the barrel when it is operated by hand.

If capstan bars *24* are to be used to operate the capstan, inserted pins *14* must be removed and pawls *15* lowered to the surface *25* of the gear housing. The pawls will engage the pockets *26* and thereby prevent reverse rotation of the barrel and cable lifter.

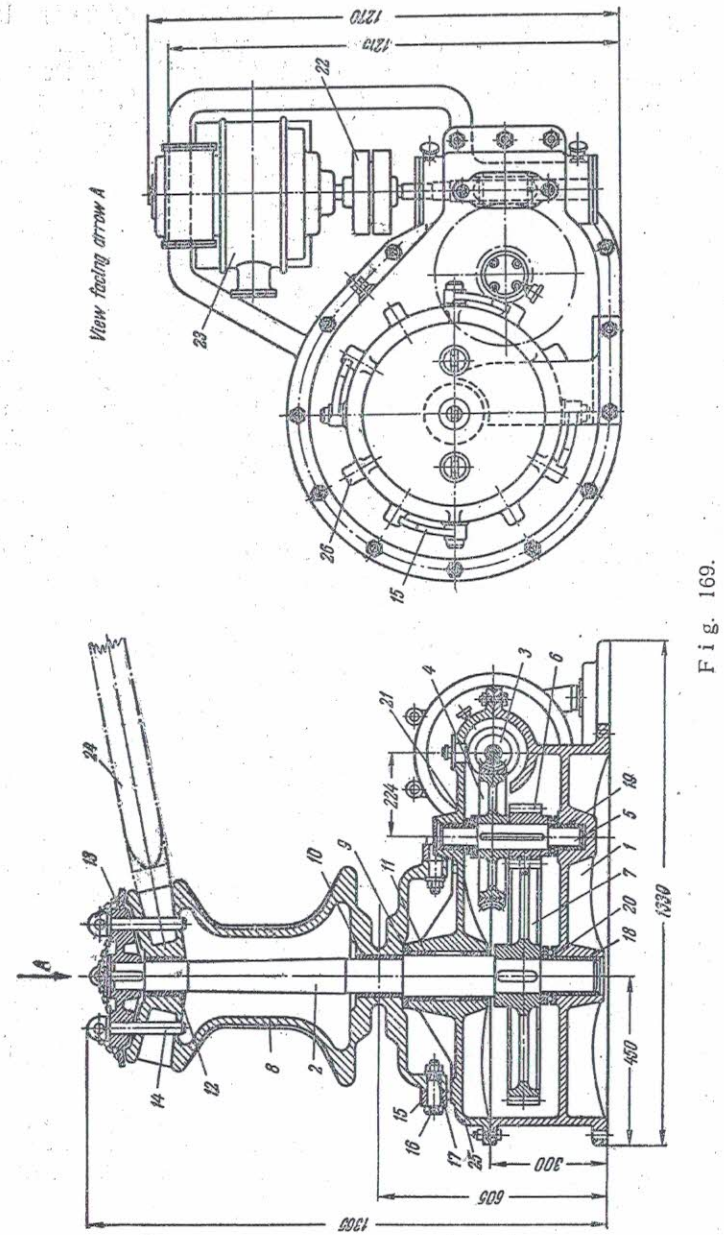


Fig. 169.

5-3. Determining the Principal Dimensions of Anchoring and Warping Machinery

The initial data used to determine the principal dimensions of anchoring machinery are the required pull of the cable lifter and the speed at which the anchor is weighed from the anchorage depth, which is equal to the distance from the hawse hole to the bottom.

It is advisable to determine the pull on the cable lifter so as to ensure that one anchor will be brought in at a speed of at least 12 m per min from the anchorage depth which is taken equal to:

80 m if each anchor weighs 1,000 kg or less
90 m if the anchor weighs from 1,500 to 3,000 kg
100 m if the anchor weighs from 3,000 to 6,000 kg.

The following notation will be used to derive the formulas for determining the pull on the cable lifter:

G_a = weight of the anchor, kg
p_a = weight per running metre of the chain cable, kg
L_a = length of the suspended cable, m
$\gamma_a = 7{,}750$ = density of the material of the anchor, kg per cu m
$\gamma_w = 1{,}025$ = density of sea water, kg per cu m
$f_h = 1.28$ to 1.35 = a factor taking into account the friction losses in the hawse hole and stopper.

The required pull of the cable lifter to hoist two anchors is

$$T_{cl} = 2f_h(G_a + p_aL_a)\left(1-\frac{\gamma_w}{\gamma_a}\right) = 2\times 1.35\,(G_a + p_aL_a)\left(1-\frac{1{,}025}{7{,}750}\right) = 2.35\,(G_a + p_aL_a) \quad \text{kg} \tag{383}$$

In hoisting one anchor

$$T_{cl} = 1.175\,(G_a + p_aL_a) \quad \text{kg}$$

The following empirical formulas can be derived from a comparison of the weights of anchors and the size of their chains as stipulated by the U.S.S.R. Shipping Register, as well as the U.S.S.R. Standard on anchor chain:

The chain bar size $d_c \approx \sqrt{G_a}$ mm. The weight per running metre of anchor chain is

$$\left.\begin{aligned} &\text{(a) } p_{ao} = 0.023\,d_c^2 \quad \text{kg for open-link chain} \\ &\text{(b) } p_{as} = 0.0218\,d_c^2 \quad \text{kg for stud-link chain} \end{aligned}\right\} \tag{384}$$

According to the U.S.S.R. Shipping Register the aft anchoring arrangement, usually consisting of a capstan, must break away the anchor and heave it in at a speed of at least 9 m per min.*

* In breaking away one anchor from the bottom

$$T_{cl} = 2G_a + 1.175\,(G_a + p_aL_a) \quad \text{kg}$$

If a windlass serves both for handling the anchor and for warping, the pull of the warp ends must not exceed

$$T_w = \frac{R_{br}}{6} \tag{385}$$

where R_{br} = breaking strength of the warping hawser.

The speed at which a capstan barrel heaves in a warping hawser can be taken from Table 58 which has been compiled from the manufacturing specifications for capstans worked out by the Central Marine Research Institute of the U.S.S.R.

Table 58

Pull of the capstan barrel, kg	Hawser heaving-in speed, m per sec	Useful power, kg-m/sec
1,200	0.3	360
3,000	0.25	750
4,500	0.2	900
7,000	0.167	1,165
12,000	0.150	1,800

The speed at which a warping hawser is heaved in by a windlass is not limited by the values in Table 58, and usually is equal to about 0.4 m per sec.

It has been stated previously that the same machinery is employed both for anchoring and warping purposes. It follows that windlasses and capstans must be designed so as to ensure normal operation of both the anchoring and warping arrangements.

As a rule, anchoring and warping capstans and windlasses are designed to ensure the proper operation of the anchoring arrangement, and then a check is made to see whether they provide for the required pull and heaving-in speed of the warping hawsers.

The number of anchors, their weight, the size of the anchor chain cables, the circumference of warping hawsers and towing ropes, and their length are determined from the tables of the pertinent regulations of the Shipping Register. To find these values it is necessary to calculate the rigging characteristic of the anchoring and warping arrangement:

$$X = L(B + H) + \Sigma\chi_i \tag{386}$$

where L = length of the ship at the summer load line, m

B = maximum breadth between the outer edges of the ship's hull, m

H = height of the side amidships, measured from the upper edge of the keel to the lower edge of the strength deck stringer, m

$\Sigma\chi_i$ = correction factor taking into account the sail effect of the superstructures.

Correction factors for the sail effect of the superstructures having a height h_i and length l_i consist of:

(a) correction factors for the superstructures of the forecastle, poop and midships, each having a length l_{sp} and height h_{sp}:

$$\chi_{sp} = k_{1sp}\Sigma l_{sp}h_{sp}$$

where k_{1sp} = 0.75 if the total length of the superstructures is equal to or less than 0.5 L

$k_{1sp} = 1.5\frac{l_{sp}}{L}$ if the total length of the superstructures exceeds 0.5 L

(b) correction factors for the deck houses, each having a height h_{dh} and length l_{dh}:

$$\chi_{dh} = k_{dh}\Sigma l_{dh}h_{dh}$$

where k_{dh} = 0.5 if the deck house has a length l_{dh} equal to or less than 0.5 L

$k_{dh} = \frac{l_{dh}}{L}$ if the length, l_{dh}, of the deck house exceeds 0.5 L.

If the breadth, b_{dh}, of the deck house exceeds its length, l_{dh}, then the product $b_{dh}h_{dh}$ is substituted into the equation in place of $l_{dh}h_{dh}$. Thus

$$\chi_{dh} = k_{dh}\Sigma b_{dh}h_{dh}$$

(c) correction factor for the quarter deck having a length l_q and height h_q:

$$\chi_q = l_q h_q$$

Data on the anchoring and warping arrangements are listed in Tables 59 and 60. The weight of each anchor is found by dividing their total weight by the number of anchors. The separate anchors may be lighter than the specified values by 7.5 per cent. The lengths of the anchor chain cables are given in the table on the assumption that the average length of each shot is 25 m. The cable length does not include the lengths of the chain slip, joining shackles, connecting links and short pieces of shots with swivels. If the tabular cable length comprises an odd number of shots, then the length of the starboard anchor chain cable is taken one shot longer than the port cable.

A section taken through the central plane of the usual five-snug cable lifter (Fig. 170) perpendicular to the shaft will be a regular

Table 59

Self-Propelled Transport Ships with an Unlimited Region of Navigation

No.	Characteristic X	Anchors: Bower: Quantity	Anchors: Bower: Total weight, kg	Anchors: Stream anchor, kg	Chain cable for bower anchors: Total length of two cables, m	Chain cable for bower anchors: Anchor chain size, mm	Chain or steel rope for the stream anchor: Length, m	Chain or steel rope for the stream anchor: Anchor chain size, mm	Chain or steel rope for the stream anchor: Diameter of steel rope, mm
1	50	2	150	25	100	12	50	—	8.8
2	75	2	200	25	125	13	50	—	8.8
3	100	2	250	50	125	15	50	—	11
4	150	2	300	50	150	16	50	—	11
5	200	2	350	50	175	17	75	—	11
6	250	2	450	75	200	18	75	11	13
7	300	2	500	75	225	19	75	13	13
8	350	2	600	100	250	20	75	14	15.5
9	400	2	700	100	275	21	75	14	15.5
10	450	2	750	125	300	22	100	15	17.5
11	500	2	800	150	300	24	100	16	17.5
12	550	2	900	175	325	25	100	16	17.5
13	600	3	1500	200	350	27	100	17	17.5
14	650	3	1700	225	350	28	100	18	19.5
15	700	3	1800	250	375	29	100	18	20.5
16	750	3	2100	250	375	30	100	19	20.5
17	800	3	2250	250	375	31	125	19	20.5
18	850	3	2400	275	375	32	125	20	22
19	900	3	2700	300	375	33	125	21	24
20	950	3	3000	300	400	34	125	21	24
21	1000	3	3200	350	400	36	125	22	24
22	1100	3	3500	400	400	37	125	23	26
23	1200	3	3750	400	420	38	150	25	26
24	1300	3	4100	450	450	40	150	25	28
25	1400	3	4250	450	450	41	150	25	28
26	1500	3	4500	500	450	42	150	26	28
27	1600	3	4750	500	450	43	150	26	28
28	1700	3	5250	600	450	45	150	28	30
29	1850	3	5500	600	450	46	150	28	30
30	2000	3	5750	700	450	46	150	29	31.5
31	2150	3	6000	700	475	48	175	29	31.5
32	2300	3	6500	800	500	49	175	29	32.5
33	2500	3	6750	800	500	50	175	29	32.5
34	2700	3	7500	900	500	52	175	30	33.5

Continued

No.	Characteristic X	Anchors: Bower: Quantity	Anchors: Bower: Total weight, kg	Anchors: Stream anchor, kg	Chain cable for bower anchors: Total length of two cables, m	Chain cable for bower anchors: Anchor chain size, mm	Chain or steel rope for the stream anchor: Length, m	Chain or steel rope for the stream anchor: Anchor chain size, mm	Chain or steel rope for the stream anchor: Diameter of steel rope, mm
35	3000	3	8250	1000	500	53	200	31	33.5
36	3300	3	9000	1000	500	55	200	31	33.5
37	3600	3	9750	1250	525	57	200	33	34.5
38	3900	3	10500	1250	550	59	225	33	34.5
39	4200	3	11000	1400	550	61	225	34	37
40	4500	3	11500	1500	550	62	225	35	37
41	4800	3	12900	1650	550	65	225	36	—
42	5100	3	13500	1750	550	67	250	37	—
43	5400	3	14500	1750	575	68	250	37	—
44	5800	3	15000	2000	600	70	250	40	—
45	6200	3	15800	2000	600	72	250	40	—
46	6600	3	16300	2250	600	74	275	43	—
47	7000	3	17600	2250	600	76	275	43	—
48	7400	3	18000	2250	600	77	275	44	—
49	7800	3	19500	2500	600	80	275	46	—
50	8200	3	20300	2700	600	82	275	48	—
51	8600	3	21000	2800	600	83	275	49	—
52	9000	3	22000	3000	600	85	275	50	—
53	9500	3	23000	3000	600	87	275	50	—

Note: Two bower anchors with a total weight of at least 2/3 of the tabular value are sufficient for ships navigating in the Caspian Sea and having a characteristic of 600 or larger.

pentagon. If the bar size of the anchor chain cable is denoted as d_c mm, then the chain pitch equal to $8\,d_c$ is to be accommodated along one side AC of the pentagon. Thus, since $AB=BC=4d_c$, it is evident from triangle OBC that the effective diameter of the cable lifter is

$$D_{cl}=2R_{cl}=2\frac{4\,d_c}{\sin\alpha^\circ}\frac{8d_c}{\sin\frac{360^\circ}{10}}=13.6\,d_c\ \text{mm}=0.013d_c\ \text{m} \quad (387)$$

The length of anchor chain cable heaved in in one revolution of the cable lifter is

$$l_c=5t_c=5\times8d_c=40d_c\ \text{mm}=0.04d_c\ \text{m} \quad (388)$$

where d_c=chain bar size, mm.

Table 60

Mooring and Warping Ropes

Characteristic	Towing rope			Warping hawsers							
								Cable warps			
	Length, m	Circumference of hemp rope, mm	Diameter of steel rope, mm	Total length, m	Number of ropes	Circumference of hemp rope, mm	Diameter of steel rope, mm	Total length, m	Number of ropes	Circumference of hemp rope, mm	Diameter of steel rope, mm
50	50	75	—	50	1	65	—	—	—	—	—
75	50	90	11	50	1	65	—	—	—	—	—
100	75	90	11	75	1	65	8.5	—	—	—	—
150	75	100	12	75	1	75	9.5	—	—	—	—
200	100	100	12	100	2	75	9.5	—	—	—	—
250	100	125	15	140	2	100	12	—	—	—	—
300	110	125	15	160	2	100	12	—	—	—	—
350	110	150	17.5	160	2	100	12	—	—	—	—
400	135	150	17.5	180	2	125	15	80	1	100	12
450	135	150	17.5	180	2	125	15	80	1	100	12
500	135	150	17.5	200	2	125	15	85	1	100	12
550	135	175	19.5	200	2	125	15	85	1	100	12
600	135	175	19.5	220	2	150	17.5	90	1	100	12
650	135	175	19.5	240	2	150	17.5	90	1	100	12
700	150	200	21.5	240	2	150	17.5	90	1	100	12
750	150	200	21.5	360	4	150	17.5	90	1	125	15
800	150	200	21.5	360	4	150	17.5	90	1	125	15
850	175	200	21.5	360	4	150	17.5	90	1	125	15
900	175	225	24	360	4	175	19.5	120	2	125	15
950	175	225	24	360	4	175	19.5	120	2	125	15
1000	175	225	24	360	4	175	19.5	120	2	150	17.5
1100	175	225	24	360	4	175	19.5	140	2	150	17.5
1200	190	250	26	360	4	175	19.5	140	2	150	17.5
1300	190	250	26	400	4	200	21.5	150	2	150	17.5
1400	190	275	28	400	4	200	21.5	150	2	150	17.5
1500	190	275	28	480	4	200	21.5	150	2	150	17.5
1600	200	300	30	480	4	200	21.5	180	2	150	17.5
1700	200	300	30	480	4	200	21.5	180	2	150	17.5
1850	200	325	32.5	540	4	200	21.5	180	2	175	19.5
2000	200	350	34.5	540	4	200	21.5	180	2	175	19.5
2150	200	350	34.5	540	4	200	21.5	180	2	175	19.5
2300	220	350	34.5	540	4	225	24	180	2	175	19.5
2500	220	350	34.5	640	4	225	24	200	2	175	19.5

Continued

Characteristic	Towing rope			Warping hawsers							
								Cable warps			
	Length, m	Circumference of hemp rope, mm	Diameter of steel rope, mm	Total length, m	Number of ropes	Circumference of hemp rope, mm	Diameter of steel rope, mm	Total length, m	Number of ropes	Circumference of hemp rope, mm	Diameter of steel rope, mm
2700	220	350	34.5	640	4	225	24	200	2	200	21.5
3000	220	350	34.5	640	4	225	24	200	2	200	21.5
3300	240	375	39	640	4	250	26	200	2	200	21.5
3600	240	375	39	640	4	250	26	200	2	200	21.5
3900	240	400	43.5	640	4	250	26	200	2	200	21.5
4200	240	400	43.5	640	4	250	26	200	2	225	24
4500	240	425	48.5	720	4	250	26	200	2	225	24
4800	240	425	48.5	720	4	250	26	200	2	225	24
5100	240	—	53	720	4	275	28	240	2	225	24
5400	240	—	53	800	4	275	28	240	2	250	26
5800	240	—	53	880	4	275	28	240	2	250	26
6200	240	—	57	960	6	300	30	240	2	250	26
6600	240	—	57	960	6	300	30	240	2	250	26
7000	240	—	57	960	6	300	30	240	2	250	26
7400	240	—	57	960	6	300	30	480	4	250	26
7800	240	—	57	960	6	300	30	480	4	250	26
8200	240	—	61.5	960	6	300	30	480	4	250	26
8600	240	—	61.5	960	6	325	32	480	4	250	26
9000	240	—	61.5	960	6	325	32	480	4	250	26
9600	240	—	61.5	960	6	325	32	480	4	250	26

Notes: 1. If the actual characteristic is between two tabular values, data should be taken for the next larger tabular characteristic.

2. The diameter and circumference of ropes selected from the table for ships with square rigging are to be increased by one size.

3. The towing rope for nonpropelling vessels is taken one size larger than the tabular value (in diameter and circumference). In addition to the towing rope indicated in the table, towing vessels (tugs) must have a towing rope for towing other vessels. This latter is to be selected in accordance with the pulling capacity of the hook which is taken with a fivefold margin of safety.

4. If Manilla or sisal hemp ropes are to be used instead of ordinary hemp, they can be taken one size less than the tabular value.

Denoting the heaving-in speed of the anchor cable as v_a m per sec, we can find the speed, n_{cl}, in rpm, of the cable lifter from the equation

$$l_c n_{cl} = 60\, v_a$$

(a) for windlasses and capstans of bower anchors:

$$n_{cl} = \frac{60\, v_a}{0.04\, d_c} = \frac{60 \times 0.2}{0.04\, d_c} = \frac{300}{d_c} \quad \text{rpm}$$

(b) for the stern anchoring capstan:

$$n_{cl} = \frac{9}{0.04\, d_c} = \frac{225}{d_c} \quad \text{rpm}$$

The efficiency of the anchoring arrangement is $\eta_a = 0.7$ to 0.85 for mechanisms with spur gearing and $\eta_a = 0.65$ to 0.75 for mechanisms

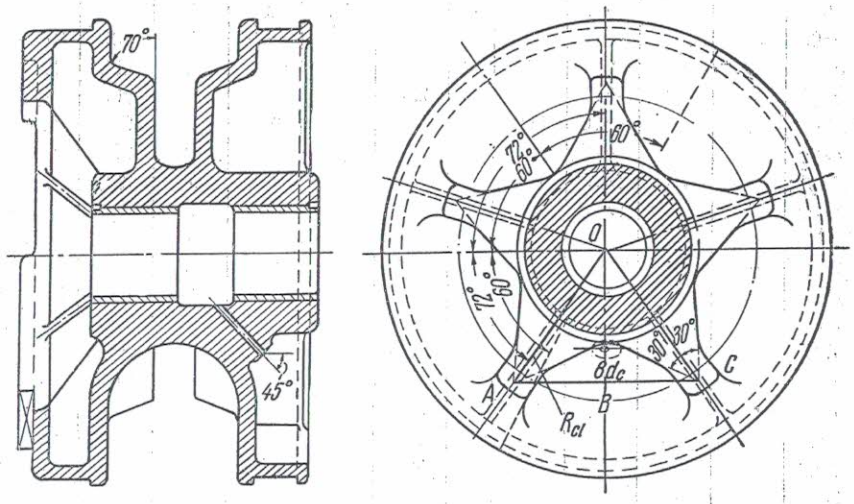


Fig. 170.

with worm gearing. It is the product of the efficiencies of the transmissions and shaft bearings in the gear train (Fig. 171):

$$\eta_a = \eta_{cl} \eta_{sh}^a \eta_{pg}^c \eta_{wg}$$

where η_{cl}, η_{sh}, η_{pg}, η_{wg} = efficiencies of the cable lifter, shaft bearings, pairs of spur gears and worm gearing

a and c = number of shaft bearings and pairs of spur gears.

The torque on the cable lifter is

$$M_{cl} = \frac{T_{cl} D_{cl}}{2\eta_{cl}} \quad \text{kg-m}$$

where $\eta_{cl} = 0.9$ to 0.92 = efficiency of the cable lifter.

Denoting the engine shaft speed as n_m, the gearing ratio of the mechanism (Table 61) is

$$i_a = \frac{n_m}{n_{cl}}$$

Table 61

Anchor handling gear	Motive unit shaft speed n_m, rpm	Gearing ratio of mechanism, i_a
Hand-operated capstans . . .		4 to 40
Steam capstans	180 to 320	18 to 60
Electric capstans	800 to 1450	110 to 200
Hand-powered windlasses . . .		9 to 18
Steam windlasses	90 to 270	6 to 30
Electric windlasses	720 to 1550	105 to 250

The torque developed on the shaft of the motive unit is

$$M_m = \frac{M_{cl}}{i_a \eta_a} \quad \text{kg-m}$$

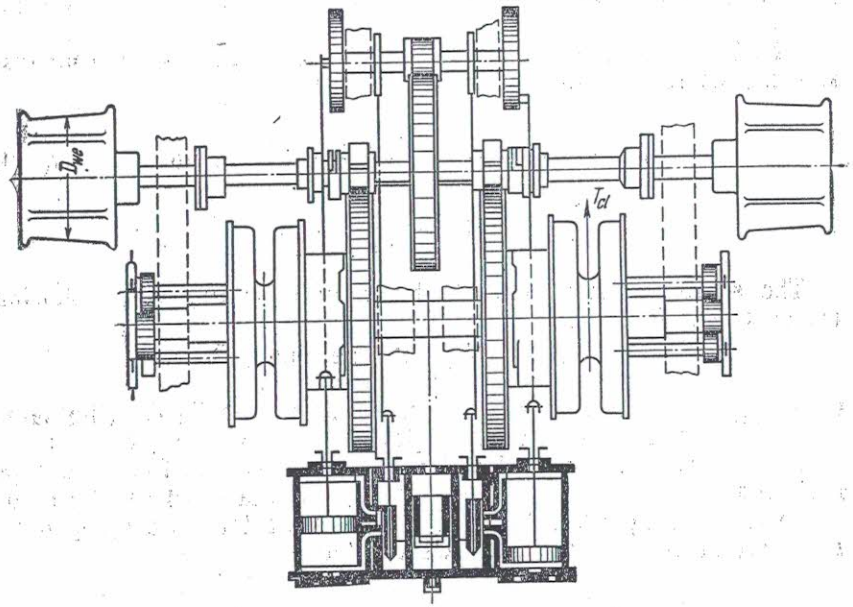


Fig. 171.

The mean shaft power of the motive unit should be

$$N_e = \frac{M_m n_m}{716.20} \quad \text{metric hp}$$

The mean indicated power is

$$N_{im} = \frac{N_e}{\eta_m}$$

The cylinder diameter of the steam engine, according to Posdyunin's formula which is based on the conditions for starting from a dead stop, is

$$D_{ca} = 1.37 \sqrt[3]{\frac{M_m}{\psi_a \eta_m (\alpha_i k_i p_{ts} - p_{ss})}} \quad \text{cm} \qquad (389)$$

where M_m = torque developed on the shaft of the engine, kg-cm
ψ_a = 0.85 to 1.7 = cylinder ratio, i.e., $S : D_{ca}$.

The value of $(\alpha_i k_i p_{ts} - p_{ss})$ is approximately from 10 to 15 per cent lower than that taken for a steering engine, due to longer distance from the anchoring mechanism to the steam supply, resulting in higher condensation losses in the pipelines. The other values in the formula are to be within the same limits as for steam steering engines.

The indicated power N_{ia} required to start the engine from rest and the coefficient of reserve power are

$$N_{ia} = \frac{\psi_a D_{ca}^3 (\alpha_i k_i p_{ts} - p_{ss}) n_m}{143{,}300} \quad \text{metric hp} \qquad (390)$$

$$\varphi_{res} = \frac{N_{ia}}{N_{im}}$$

The steam consumption of the engine driving the anchoring arrangement is

$$G_{ia} = g_{ia} N_{ia} \quad \text{kg per hour}$$

where g_{ia} = specific steam consumption, kg per ihp-hr (the same values are taken as for a steam steering engine).

If need arises to determine the pull on the cable lifter from data measured on the anchoring mechanism, formula (390) can be used.

Solving Posdyunin's formula (389) for the torque developed on the shaft of the steam engine we can write

$$M_m = \left(\frac{D_{ca}}{1.37}\right)^3 \eta_m \psi_a (\alpha_i k_i p_{ts} - p_{ss}) \quad \text{kg-cm}$$

On the other hand, if i_a is the total gearing ratio of the transmission in the anchoring mechanism, then

$$M_m = \frac{M_{cl}}{\eta_a i_a} = \frac{T_{cl} D_{cl}}{2\eta_a i_a} \quad \text{kg-cm}$$

Combining the last two equations we obtain

$$T_{cl} = \frac{2M_m \eta_a i_a}{D_{cl}} = 2\left(\frac{D_{ca}}{1.37}\right)^3 \frac{\eta_m \psi_a (\alpha_i k_i p_{ts} - p_{ss}) \eta_a i_a}{D_{cl}} =$$

$$= 0.78 \frac{D_{ca}^3}{D_{cl}} \eta_m \psi_a (\alpha_i k_i p_{ts} - p_{ss}) \eta_a i_a \quad \text{kg}$$

The diameter of the warp ends is taken equal to

(a) $D_{we} = (5 \text{ to } 8)\, d_w$ for hemp ropes (391)

(b) $D_{we} = (15 \text{ to } 20)\, d_w$ for steel ropes (392)

where d_w = diameter of the warping hawser.

Warp end diameters determined from the diameter of hemp ropes will be suitable for steel ropes as well.

Denoting the hawser heaving-in speed as v_w m per sec we can find the speed of the warping shaft from the length of hawser heaved in per minute. Thus

$$n_w = \frac{60 v_w}{\pi (D_{we} + d_w)} = 19.1 \frac{v_w}{D_{we} + d_w} \quad \text{rpm} \qquad (393)$$

where v_w = hawser heaving-in speed, m per sec, is to be assigned according to the pull of the warp end (Table 58).

The gearing ratio between the warping shaft and the shaft of the motive unit is

$$i_w = \frac{n_m}{n_w}$$

The pulling force developed on the warp end is

$$T_{we} = \frac{M_{we}}{\frac{1}{2}(D_{we} + d_w)} = \frac{2M_m \eta_w i_w}{D_{we} + d_w} \leqslant \frac{R_{br}}{6} \qquad (394)$$

where M_{we} = torque developed on the warp end
η_w = efficiency of the transmission between the warping and motive unit shafts.

If n_m rpm is the speed of the motive unit shaft, the speed at which the hawser is heaved in will be

$$v_w = \frac{\pi (D_{we} + d_w) n_m}{60 i_w} \quad \text{m per sec} \qquad (395)$$

The pulling force and heaving-in speed obtained from the above formulas must conform to the requirements for warping appliances. If the heaving-in speed is unsuitable, it is necessary to change the gearing ratio of the transmission between the motive unit shaft and the warping shaft correspondingly. If the mechanism does not develop the required pulling force on the warp ends, then its power requirements must be determined from the conditions stipulated for warping appliances.

The gearing ratio required to develop a pulling force T_{we} for a given torque on the motive unit shaft can be found from the following formula:

$$i_w = \frac{T_{we}(D_{we}+d_w)}{2M_m\eta_w} \tag{396}$$

The power ratings of the electric motors for driving anchoring arrangements and their shaft speeds are selected from nominal values in the same way as for a steering gear.

The gearing ratio of the mechanism is found from the formula:

$$i_a = \frac{n_r \pi D_{cl}}{60 v_a} \tag{397}$$

The rated torque of the motor is

$$M_r = \frac{T_{cl}D_{cl}}{2i_a\eta_a} \quad \text{kg-cm} \tag{398}$$

where T_{cl} = required pull on the cable lifter
η_a = overall efficiency of the mechanism.

The mean effective power of the motor is

$$N_e = \frac{M_r n_r}{97{,}500} \quad \text{kW} \tag{399}$$

The regulations of the U.S.S.R. Shipping Register stipulate that each anchoring and warping arrangement must have an emergency hand drive. A rocker bar usually serves as the emergency hand drive for windlasses. The power requirements of this mechanism are calculated on the basis of the force required to break away and hoist one anchor.

The pull required to hoist the anchor, for the hand drive shown schematically in Fig. 172, is determined from the condition of equality of the moments of acting forces in reference to the main shaft.

The moment, developed by the reaction of the anchor and counteracting the effort to break it away, is

$$M_{ar} = \frac{T_{cl}D_{cl}}{2\eta_{cl}} \quad \text{kg-m} \tag{400}$$

where η_{cl} = 0.94 = efficiency of the cable lifter.

The moment developed by the forces applied to the ends of the rocker arm is

$$M_{ar} = P_{tr}(R_{rw}+a)\eta_{rw} \quad \text{kg-m} \tag{401}$$

where R_{rw} = 0.3 to 0.8 = radius of the ratchet wheel, m
η_{rw} = 0.92 = efficiency value accounting for the friction in the teeth of the ratchet wheel and in the joints of the tie rod and shoe
a = 0.13 to 0.21 = distance from the circumference of the ratchet wheel to the tie rod axis, m
P_{tr} = force acting on the tie rod, in kg.

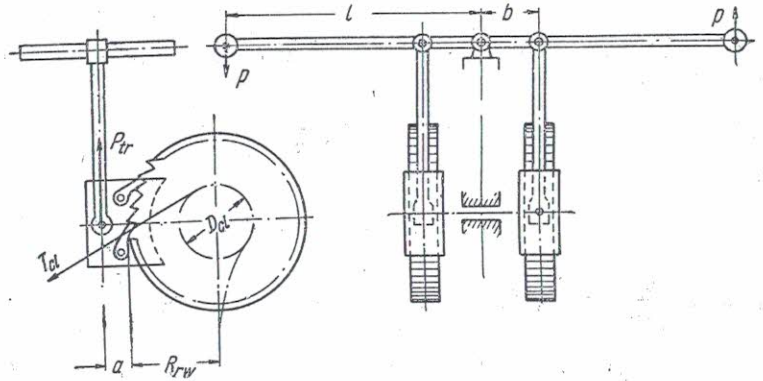


Fig. 172.

Combining equations (400) and (401) we obtain

$$P_{tr} = \frac{T_{cl}D_{cl}}{2(R_{rw}+a)\eta_{rw}\eta_{cl}} \quad \text{kg} \tag{402}$$

The force applied to the tie rod when the rocker arm is operated is found from the equation for the moments of all forces applied to the rocker arm in reference to its pivot:

$$p_{dh}k_{dh}l\eta_{hj} = P_{tr}b$$

whence

$$p_{dh} = \frac{P_{tr}b}{k_{dh}l\eta_{hj}} \tag{403}$$

where p_{dh} = force exerted by one deck hand operating the rocker arm, kg
k_{dh}=4 to 16 = number of deck hands operating the rocker arm
η_{hj}=0.98 = efficiency of the hinged joints (including the pivot) of the rocker arm

l and b = arm lengths at which forces p_{dh} and P_{tr}, respectively, are applied as measured from the pivot of the rocker arm.

Combining equations (402) and (403) we find that the force exerted by one deck hand operating the rocker arm is

$$p_{dh}=\frac{T_{cl}D_{cl}}{2k_{dh}(R_{rw}+a)\,\eta_{rw}\eta_{cl}\eta_{hj}}\frac{b}{l} \qquad (404)$$

The force p_{dh} exerted by each deck hand in breaking away the anchor may be taken equal to 25 or 30 kg while that in hoisting the anchor should not exceed 16 kg. The ratio of the arms $b : l$ ranges from 0.2 to 0.07 while the total length $2l$ may reach 7 m.

Next we shall determine the speed at which the anchor is hoisted. Denoting the speed of tie rod movement as v_{tr} we can write the following equation which is sufficiently accurate for practical purposes:

$$\frac{v_a}{v_{tr}}=\frac{D_{cl}}{2(R_{rw}+a)}$$

then

$$v_a=v_{tr}\frac{D_{cl}}{2(R_{rw}+a)}$$

But, since $\frac{v_{ra}}{v_{tr}}=\frac{l}{b}$, then $v_{tr}=v_{ra}\frac{b}{l}$. Therefore

$$v_a=v_{tr}\frac{D_{cl}}{2(R_{rw}+a)}=v_{ra}\frac{D_{cl}}{2(R_{rw}+a)}\frac{b}{l}\ \text{m per sec} \qquad (405)$$

where v_{ra} = 0.5 to 0.8 = speed of rocker arm motion, m per sec.

As a rule, the anchor is hoisted with a hand drive at a speed ranging from 2.5 to 3 m per sec.

If the warp ends are mounted on the same shaft as the ratchet wheel, the speed at which the hawsers are heaved in can be determined from equation (405) except that D_{we} is substituted for D_{cl}.

Capstan bars or spur and worm gearing operated by crank handles are commonly used as hand drives for capstans. The initial equation in determining the force that must be applied to the capstan bars to operate the capstan (Fig. 173) will be the equality of the moments developed by the reaction of the anchor and the forces which break it away:

$$p_{dh}Lk_{dh}=\frac{T_{cl}D_{cl}}{2\eta_d} \qquad (406)$$

where L = 2 to 3 = nominal length of the capstan bars as measured from the capstan shaft to the point where the force is applied, m

η_d = 0.9 to 0.92 = overall efficiency of the drive taking into consideration the friction losses in the cable lifter and the bearings of the warping barrel

k_{dh} = 4 to 12 = number of deck hands operating the capstan.

The force exerted by one deck hand is

$$p_{dh}=\frac{T_{cl}D_{cl}}{2Lk_{dh}\eta_d} \qquad (407)$$

At the moment the anchor is broken away the force p_{dh} exerted by one deck hand may reach 40 kg. The force p_{dh} should not exceed 16 kg when the anchor cable is being brought in.

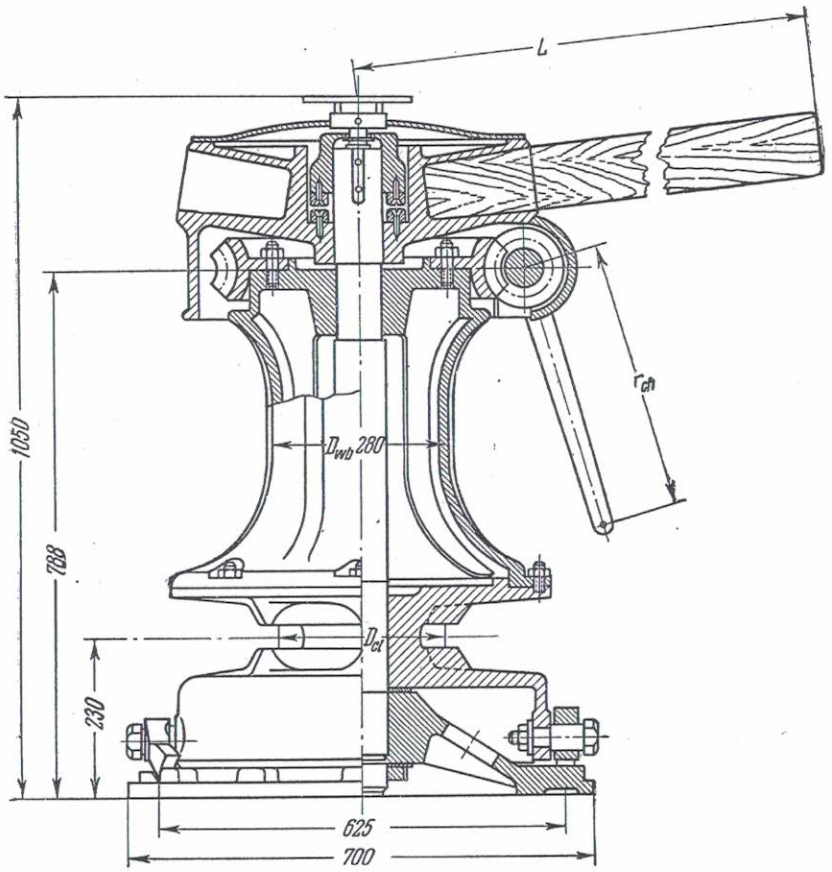


Fig. 173.

The speed at which the anchor is hoisted

$$v_a = v_{cb} \frac{D_{cl}}{2L} \tag{408}$$

where v_{cb} = 0.4 to 0.5 = circumferential speed of the capstan bars, m per sec.

The equality of moments is expressed in the following manner for capstans with spur and worm gear drives:

$$r_{ch} p_{dh} k_{dh} i_g \eta_g = \frac{T_{cl} D_{cl}}{2\eta_{cl}} \tag{409}$$

where r_{ch} = radius of the crank handle, m

i_g = gearing ratio of the installed transmission (i_g = = 30 to 40 for worm gearing and i_g = 4 to 6 for spur gearing)

η_g = efficiency of this transmission with allowance for the friction losses in the bearings of the driving shaft.

The force exerted by one deck hand is

$$p_{dh} = \frac{T_{cl} D_{cl}}{2 r_{ch} k_{dh} i_g \eta_g \eta_{cl}} \quad \text{kg} \tag{410}$$

The anchor hoisting speed is

$$v_a = v_{ch} \frac{D_{cl}}{2 r_{ch} i_g} \quad \text{m per sec} \tag{411}$$

where v_{ch} = 0.6 to 0.8 = circumferential speed of the crank handles, m per sec.

These formulas can also be used to determine the required effort and speed at which hawsers are heaved in if T_{wb} is substituted for T_{cl} and D_{wb} for D_{cl} (T_{wb} and D_{wb} being the pull on and the diameter of the warping barrel of the capstan). The moments of the tangential forces and the principal dimensions of the elements in the transmission may be determined by the formulas given for steering engines.

5-4. Fundamentals of Windlass and Capstan Operation

Before starting a windlass or capstan:

(1) check whether its operation is hindered by any foreign objects;

(2) apply lubricant to all points of lubrication, put all oil and grease cups into working order and check the oil level in the baths of the worm gear transmissions;

(3) open the blow-off valves of the cylinders and the valve of the live steam line;

(4) open the valves in the pipelines supplying live steam to the windlass or capstan and exhausting the spent steam;

(5) apply the band brake and disengage the cable lifters from the motive unit;

(6) check whether the clutches engage without failure;

(7) check whether the hand drive is properly disengaged;

(8) open the exhaust steam valve fully, crack the live steam valve and begin to blow through and warm up the cylinders of the windlass (or capstan) steam engine, continuing until no condensate is seen in the steam issuing from the blow-off valves;

(9) after warming up the engine make sure that the windlass or capstan is operating properly by running its shaft several revolutions in each direction. If no knocking is heard, the windlass or capstan is ready for operation.

During operation it is necessary to see that the lubricant is properly fed and to listen for knocking. If any abnormal noise is heard the windlass should be stopped to find out what the trouble is and to make the necessary repairs without delay.

When the windlass is stopped for a short time it is necessary to close the live steam valve and then the exhaust valve and to open the blow-off valves.

If the windlass or capstan is to stand idle for a prolonged period, it will be necessary, in addition to the preceding, to remove the wicks from the oilers, close the valves in the live and spent steam lines, and to test the action of the band brake and friction clutches.

Whenever a steam windlass or capstan is stopped in the winter time, the condensate must be carefully drained from the steam lines.

Operation of the windlass from the hand drive must be checked once a month.

Chapter 6

WINCHES AND CRANES

6-1. Cargo Winches and Cranes

How a ship observes its cargo carrying schedule depends to a great extent upon the length of its stand in harbour to load or discharge cargo.

Cargo is loaded and discharged with:

(1) shipboard cargo handling facilities;

(2) harbour cargo handling facilities;

(3) a combination of shipboard and harbour facilities.

It would seem at first thought that the equipment of ships with cargo handling appliances is inexpedient for the following reasons:

(1) Much loading and unloading equipment is concentrated in large harbours.

(2) The cargo carrying capacity of a ship is reduced by the weight of its cargo handling appliances.

(3) The initial cost of this equipment and its maintenance represent a considerable item of expense.

(4) The costs are not readily justified because the equipment operates only a short time in comparison with the service time of the ship.

Nevertheless, distant-cruise cargo ships must have their own handling equipment because:

(1) on distant cruises ships may call at ports that are poorly equipped with harbour mechanization and cargo handling devices;

(2) it is sometimes necessary to load or discharge cargo in a roadstead or even at sea;

(3) up-to-date high-speed techniques for loading and discharging cargo in conjunction with skilful utilization of shipboard and harbour handling equipment considerably reduce the time required in harbour;

(4) the harbour facilities may prove incapable of simultaneously loading a large number of ships that may happen to be in harbour at the same time.

The cargo handling equipment of a ship consists of the hatches, derricks with their rigging operated by winches, and cargo cranes.

Cargo cranes are particularly convenient for ships which perform handling operations in a roadstead or for liners with frequent ports of call and short stands. The convenience of using cranes instead of derricks, especially on passenger and passenger-cargo carriers, is due to the fact that cranes are always ready for action while derricks must be rigged each time before calling at a port.

The passenger vessels navigating the Crimean-Caucasian Line can serve as an example.

Winches and cranes must:

(1) ensure a high load-handling capacity;

(2) be dependable in operation;

(3) be convenient in operation, maintenance and repairs;

(4) be highly economical and have a small overall size and weight.

(A) WINCHES

In accordance with the manner in which they are powered, winches are classified as hand, steam, electric, hydraulic or engine driven. Steam and electric winches are the most widely employed; the former due to their simple construction, low cost and reliable operation, and the latter because of the low energy losses in transmission lines, simple maintenance and repairs and high economy in operation.

Hydraulic winches, operating upon the same principle as hydraulic steering engines, have not found wide application because of their complex construction and low economy though they provide smoother adjustment of the hoisting speed, a wider speed range and are silent in operation.

Winches powered by internal combustion engines are rarely used because of their complex construction, high cost, incapability of starting under load, narrow speed range and difficulty in varying their speed, even though they are highly economical. At present, such winches are installed only on sailing ships which do not have an adequate power plant.

Hand winches are employed only on small vessels. Their lifting capacity does not exceed 1.5 tons at a hoisting speed of from 1 to 1.5 m per minute. Hand winches are usually operated by 3 or 4 deck hands. They may be of various design but the principle by which a load of definite weight is handled at the required hoisting speeds is the same and is based upon the selection of the proper spur and worm gearing.

Steam and electric winches may incorporate either one or two transmissions. In the latter case, the single transmission is used to handle the lighter loads at a higher hoisting speed (from 0.5 to 1 m per sec).

The hoisting speed can be lowered to 0.25 or 0.5 m per sec by engaging the double transmission. This enables the lifting capacity to be increased without increasing the power of the motive unit.

One arrangement for handling cargo is shown in Fig. 174. The end *2* of boom *1* is linked through a swivel joint (heel shackle) to the heel shoe which is fastened to the mast. From its initial position, shown in the upper drawing with dot-and-dash lines, the boom is raised by tackle *4* into the working position, at a height providing the required reach of the boom, and is secured to the mast with topping lift *3*.

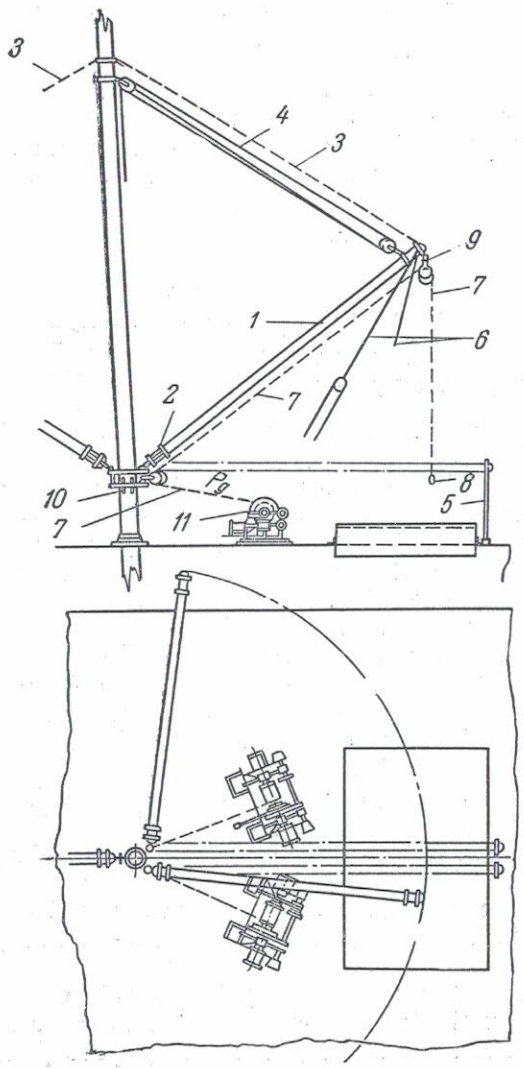


Fig. 174.

When the ship is under way the boom is secured in rest *5*. The boom is swung to transfer the hoisted load by means of the guys *6*. Cargo whip *7*, ending in hook *8*, passes over blocks *9* and *10* and runs to the barrel of winch *11* which performs the handling operations.

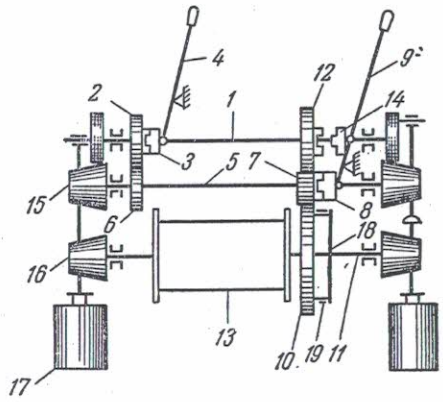


Fig. 175.

Winches may be of either the single-or two-speed type and may have either one or two barrels. Two-speed single-barrel winches are the most extensively used type.

The two-speed steam cargo winch shown schematically in Fig. 175 can perform three kinds of operations.

The winch is driven by a two-cylinder steam engine *17*. This steam engine can operate both with and without steam expansion. Gear *2*, mounted freely on the driving shaft *1*, has a jaw-type clutch member. Gear *12*, also with a clutch member, runs freely on the same shaft. Besides these, clutch members *3* and *14* slide along feathers on shaft *1*. Gear *2* meshes with gear *6* which is keyed on shaft *5* while *12* meshes with gear *7*, having a clutch member and mounted freely on the same shaft. Gear *7*, in turn, meshes with gear *10* which is fixed on load shaft *11* as is the winch barrel *13*. Winch heads *15* and *16* are mounted at the ends of shafts *5* and *11*. When levers *4* and *9*, and clutch members *3*, *8* and *14* are shifted to the positions shown in Fig. 175, we have the double-gear arrangement and shafts *5* and *11* with barrel *13* and winch heads *15* and *16* are driven from shaft *1* through gears *2*, *6*, *7* and *10*.

If clutch members *3* and *8* are disengaged and member *14* is engaged, then, through gears *12*, *7* and *10*, shaft *1* will drive the load shaft *11* together with barrel *13* and winch heads *16*. This is the single-gear arrangement of the winch.

When clutch member *3* is engaged and clutch members *14* and *8* are disengaged, only shaft *5* with winch heads *15* will operate. Barrel *13* has a drum *18* on which a brake band *19* is applied by a hand lever or pedal for braking purposes. The engines of steam winches operate both with or without steam expansion and with steam admission during 70 to 100 per cent of the stroke. Winch engines operating with steam expansion are usually reversed by a Stephenson linkage while those operating without expansion are reversed by a starting valve.

Electric winches may be designed with either spur or worm gearing.

The electric winch illustrated in Fig. 176 is designed with spur gearing to increase its efficiency. Such gearing, however, increases the overall size of the winch as compared to winches with worm gearing, makes it more difficult to obtain a compact arrangement of the parts and is the source of considerable noise in operation.

In this winch, supports *2* of the shaft bearings are secured to baseplate *1* with studs and nuts. The winch is powered by electric motor *6* which is controlled from handwheel *7* of the controller. The motor shaft is linked through flexible coupling *8* with the shaft of the gearbox. The latter is enclosed in housing *9* and consists of a pinion and a gear. Pinion *10* is keyed on the shaft of the gear and meshes with gear *11* which is of welded construction. This gear is fixed on load shaft *5*. Winch heads *4* are keyed, clutch member *12* slides along a feather and barrel *3* is mounted freely on the load shaft. The barrel is cast integral with disk *13* of the foot brake. The end faces of clutch member *12* and the hub of disk *13* are made with jaws. Clutch member *12* can be shifted by lever *14* into engagement with the hub of disk *13*; this links barrel *3* to shaft *5*. The foot brake is applied through a lever system from pedal *18*.

The winch also has an electromagnetic brake *15* which operates automatically. As long as there is current in the motor circuit, the electromagnetic brake keeps band *16* on the disk of coupling *8* in the released position. At a power failure, the brake band is applied on the disk by the weight *17*, thereby stopping the winch.

To perform cargo handling operations clutch member *12* must be in engagement with the hub of disk *13* and, therefore, with barrel *3*. In this case, shaft *5*, barrel *3* and winch heads *4* will be driven from the motor through the gearing. If rotation of the barrel is not required for the job to be done, clutch member *12* is disengaged. Evidently, only the winch heads *4* will now be driven by the motor.

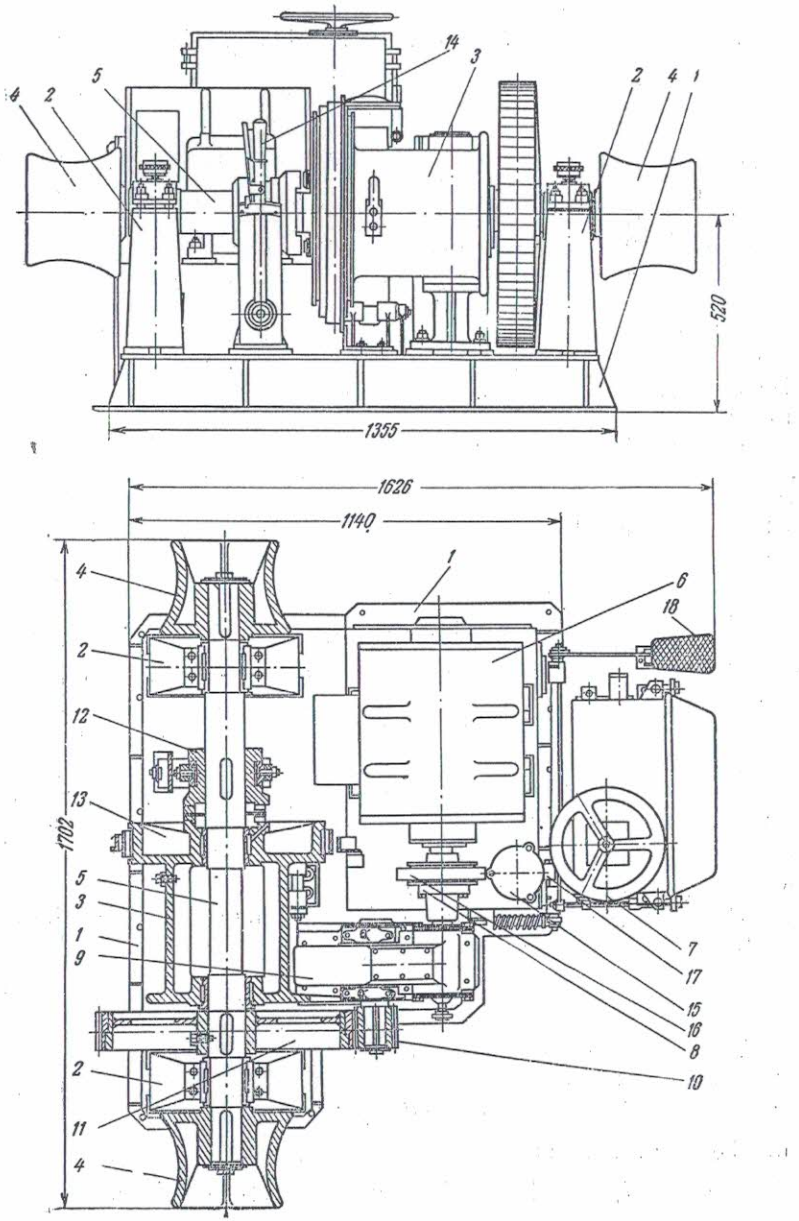


Fig. 176.

The pulling force, T_b, on the winch barrel (Figs. 174 and 175) is

$$T_b = \frac{P+Q}{\eta_p^k} = P_g \qquad (412)$$

where P = weight of the useful load being hoisted, kg
Q = (0.0028 to 0.0022) P = weight of the cargo hook with the shackle, kg
η_g = 0.9 to 0.96 = efficiency of one pulley
k = number of intermediate pulleys between the boom iron and the winch barrel.

A two-speed winch is designed for a given pulling force, P_g, in double-gear operation. The gearing ratio for single-gear operation usually ranges from 4 to 6 and for double-gear operation, from 6 to 12.

The diameter of the winch barrel is taken to be

$$D_b = (16.5 \text{ to } 18)\, d_r \qquad (413)$$

where the rope diameter, d_r, is selected according to its breaking strength, $R_{br} = 6P_g$.

The length of the winch barrel is

$$L_b = (1.1 \text{ to } 1.6)\, D_b \quad \text{cm}$$

The number of layers, z, of rope wound on the barrel depends upon the size of the latter and the length of the rope to be wound. This length, L_r, ranges from 40 to 75 m and the number of layers does not exceed five.

The diameter, d_r, and kind of rope is selected so that the actual tensile strength

$$R_{br} \geqslant k_m P_g \text{ kg} \qquad (414)$$

where $k_m \geqslant 6$ = margin of safety.

The number of turns that can be accommodated along the length of the barrel is

$$m = \frac{L_b}{d_r} \qquad (415)$$

The length of rope accommodated is:

in the first layer

$$l_{m1} = \pi (D_b + d_r)\, m$$

in the second layer

$$l_{m2} = \pi [D_b + (4d_r - d_r)]\, m = \pi (D_b + 3d_r)\, m$$

in the third layer

$$l_{m3} = \pi [D_b + (6d_r - d_r)]\, m = \pi (D_b + 5d_r)\, m$$

in the z-th layer

$$l_{mz} = \pi [D_b + (2zd_r - d_r)]\, m = \pi [D_b + d_r (2z-1)]\, m \qquad (416)$$

Thus, the rope capacity of the barrel, if the rope is wound, is:

in one layer

$$L_{m1} = l_{m1} = \pi (D_b + d_r)\, m$$

in two layers

$$L_{m2} = l_{m1} + l_{m2} = \pi (D_b + d_r)\, m + \pi (D_b + 3d_r)\, m = \pi (2D_b + 4d_r)\, m$$

in three layers

$$L_{m3} = L_{m2} + l_{m3} = \pi (2D_b + 4d_r)\, m + \pi (D_b + 5d_r)\, m = \pi (3D_b + 9d_r)\, m$$

in z layers

$$L_{mz} = \pi (zD_b + z^2 d_r)\, m \qquad (417)$$

The barrels of cargo winches are made with a smooth surface to increase their rope capacity, reduce their cost and to simplify their manufacture. They are not equipped with rope guiding devices. This leads to a certain lack of uniformity in the arrangement of the rope on the barrel which is taken into account by the factor $\beta_b \approx 0.9$ in calculating the rope capacity.

Thus equation (417) can be rewritten as

$$L_{mz} = \beta_b \pi m\, (zD_b + z^2 d_r)\, m \qquad (418)$$

If the required rope capacity of the winch barrel is known beforehand, equation (418) can be solved for z to find the number of layers in which the rope will be accommodated. Thus

$$\beta_b \pi m d_r z^2 + \beta_b \pi m D_b z - L_{mz} = 0$$

$$z^2 + \frac{D_b}{d_r} z - \frac{L_{mz}}{\beta_b \pi m d_r} = 0$$

then

$$z = -\frac{D_b}{2d_r} + \sqrt{\left(\frac{D_b}{2d_r}\right)^2 + \frac{L_{mz}}{\beta_b \pi m d_r}} = -\frac{D_b}{2d_r} + \sqrt{\frac{D_b^2}{4d_r^2}\left(1 + \frac{4L_{mz}d_r^2}{\pi D_b^2 \beta_b m d_r}\right)}$$

$$z = \frac{D_b}{2d_r}\left(\sqrt{1 + \frac{4d_r L_{mz}}{\beta_b D_b^2 \pi m}} - 1\right) \qquad (419)$$

The maximum torque on the winch barrel is usually developed at the beginning in hoisting the load or at the end in lowering it since it is necessary to overcome the inertia of the load. Moreover, the torque increases at the end of hoisting because the rope is wound in z layers.

In the latter case, calculations are usually conducted using the design diameter of the barrel which is

$$D_{bd} = D_b + d_r(2z-1) \quad \text{m} \tag{420}$$

The torque developed on the barrel shaft is

$$M_{bd} = \frac{1}{2}[D_b + d_r(2z-1)]\frac{T_b}{\eta_b} \quad \text{kg-m} \tag{421}$$

where η_b = efficiency of the winch barrel.

The rotational speed, n_{bd}, of the barrel is found from the following equation for a load hoisting speed v_{ld} with the double gearing of the winch engaged:

$$n_{bd} = \frac{60v_{ld}}{\pi D_{bd}} = 19.1\frac{v_{ld}}{D_{bd}} \quad \text{rpm} \tag{422}$$

The overall gearing ratio of the winch with the double gearing engaged is

$$i_{wd} = \frac{n_m}{n_{bd}} = \frac{n_m}{\frac{60v_{ld}}{\pi D_{bd}}} = \frac{n_m \pi D_{bd}}{60v_{ld}} \tag{423}$$

where n_m=80 to 250 = speed of the winch steam engine shaft, rpm
n_m = 500 to 3,000 = shaft speed of the electric motor, rpm.

The overall efficiency, η_{wd}, of the winch when the double gearing is engaged is the product of the efficiencies of the shafts (η_{sh}), pairs of spur gears (η_{pg}), barrel (η_b) and worm gearing (η_{wg}).

Thus

$$\eta_{wd} = \eta_{sh}^a \eta_{pg}^c \eta_b \eta_{wg} \tag{424}$$

where a and c = number of shafts and pairs of gears, respectively
η_{wd} = 0.7 to 0.85 for winches with spur gearing
η_{wd} = 0.65 to 0.75 for winches with worm gearing.

The required shaft torque of the motive unit is

$$M_{md} = \frac{M_{bd}}{i_{wd}\eta_{wd}} \quad \text{kg-m} \tag{425}$$

The diameter of the steam engine cylinder and the required power to start from rest are determined from Posdyunin's formula:*

$$D_{cw} = 1.37\sqrt[3]{\frac{M_{md}}{\Psi_r \eta_m (\alpha_i k_i p_{ts} - p_{ss})}} \quad \text{cm} \tag{426}$$

* The symbols denote the same values as in the case of steering engines.

where $k_i = \frac{1+\ln\Delta}{\Delta}$ = coefficient of mean theoretical indicated pressure for a ratio of steam expansion Δ
M_{md} = torque developed on the engine shaft, kg-cm.

The indicated power of the engine required to start from rest under load is

$$N_i = \frac{D_{cw}^3(\alpha_i k_i p_{ts} - p_{ss})\, n_m \Psi_r}{143{,}300} \quad \text{hp} \tag{427}$$

Values of k_i as a function of the admission ratio (reciprocal of the expansion ratio) $\delta = \frac{1}{\Delta}$ are listed in Table 62.

Table 62

δ	0.5	0.6	0.7	0.8	0.9	1
k_i	0.848	0.907	0.95	0.979	0.995	1

If T_{bs} is the given rated pulling force for the single gearing engagement of the winch, calculated from equation (412) for the given load hoisting capacity, then, according to equation (421), the torque developed on the winch barrel is

$$M_{bs} = \frac{1}{2}[D_b + d_r(2z-1)]\, T_{bs} \quad \text{kg-m} \tag{428}$$

Assuming that the motive unit shaft rotates at a constant speed n_m we can write

$$\frac{M_{bd}}{i_{wd}\eta_{wd}} = \frac{M_{bs}}{i_{ws}\eta_{ws}} \tag{429}$$

where η_{ws} = overall efficiency of the winch when the single gearing is engaged
i_{ws} = gearing ratio of the winch with the single gearing engaged.

It follows that the required gearing ratio is

$$i_{ws} = \frac{M_{bs}}{M_{bd}}\frac{\eta_{wd}}{\eta_{ws}} i_{wd} \tag{430}$$

The speed of the winch barrel for single gearing is

$$n_{bs} = \frac{n_m}{i_{ws}} \quad \text{rpm} \tag{431}$$

The load hoisting speed is

$$v_l = \frac{\pi D_{bd} n_{bs}}{60} \quad \text{m per sec}$$

In cases when the load lifting capacity of an existing winch is not known, the rated pulling force on the barrel can be determined by taking the measurements of the principal elements. Indeed, combining equations (426), (425) and (421), we obtain

$$D_{cw} = 2\sqrt[3]{\frac{M_{md}}{\pi \Psi_r \eta_m (\alpha_i k_i p_{ts} - p_{ss})}} = 2\sqrt[3]{\frac{\frac{M_{bd}}{i_{wd}\eta_{wd}}}{\pi \Psi_r \eta_m (\alpha_i k_i p_{ts} - p_{ss})}} =$$
$$= 1.37\sqrt[3]{\frac{0.5\,[D_b + d_r(2z-1)]\,T_b}{\Psi_r \eta_m (\alpha_i k_i p_{ts} - p_{ss})\, i_{wd}\eta_{wd}}}$$

then

$$T_b = \frac{D_{cw}^3 \Psi_r \eta_m (\alpha_i k_i p_{ts} - p_{ss})\, i_{wd}\eta_{wd}}{1.37^3 \times 0.5\,[D_b + d_r(2z-1)]} \approx$$
$$\approx 0.78 \frac{D_{cw}^3 \Psi_r \eta_m (\alpha_i k_i p_{ts} - p_{ss})\, i_{wd}\eta_{wd}}{D_b + d_r(2z-1)} \qquad (432)$$

The steam consumption, G_{sw}, for steam cargo winches and the electrical energy consumed, W_{ew}, by electric winches can be found from the following formulas:

$$G_{sw} = (1.5z_w + 1.2g_s)(0.54H_h + 3) \quad \text{kg per hour}$$
$$W_{ew} = (0.42z_w + 0.02g_s)(0.5H_h + 3) \quad \text{kW}$$

where z_w = number of winches in operation
g_s = scale of cargo operations, tons per hour
H_h = average height from the flooring of the double bottom to the hatch coaming, m.

Steam winches consume about 50 kg of steam in handling one ton of cargo; electric winches consume about 0.75 kWh.

Electric cargo winches can be installed with equal advantage on both steam and diesel-powered ships. The motor shaft speed ranges from 500 to 3,000 rpm while heavy loads are hoisted at speeds up to 0.7 m per sec. Electric winches usually have from 4 to 6 different speeds. Worm gearing is not being employed in winches much at present because of its low efficiency and inevitable overheating. Up-to-date winch models of Soviet manufacture are designed with spur gearing. Recent trends in this field are to accomplish winch reversal by means of consecutively linked transmissions and cone clutches, maintaining a constant direction of motor rotation, and a wider use of hydraulic winches.

(B) CRANES

Cargo cranes possess the following advantages over cargo handling appliances consisting of booms and winches:

(1) higher load-handling capacity;
(2) less personnel requirements for cargo operations;

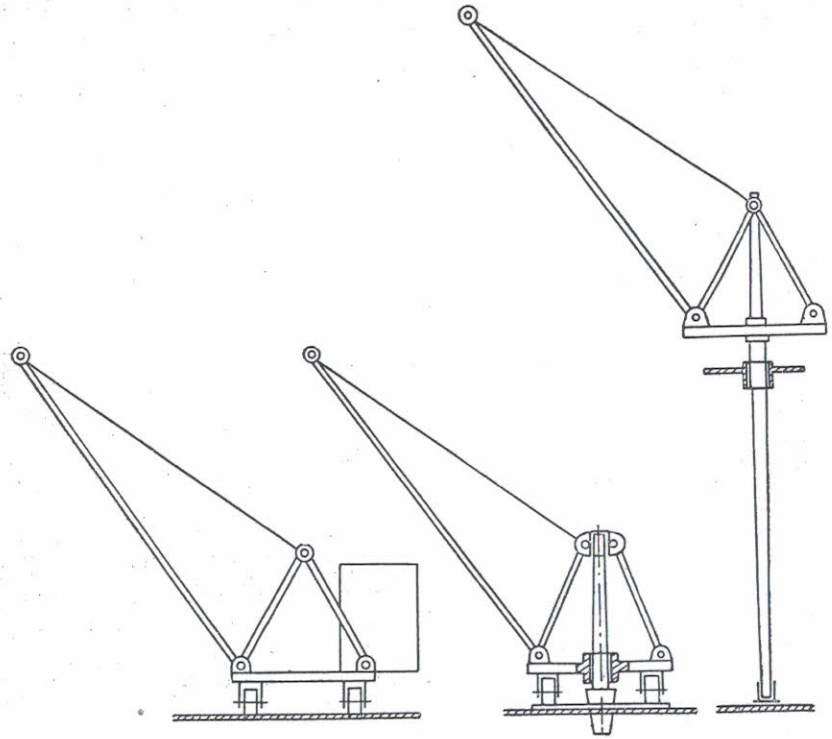


Fig. 177. Fig. 178. Fig. 179.

(3) readiness for action at all times while booms must be rigged before each call at a port, an operation requiring much time and labour;

(4) feasibility of serving two adjacent hatches with one crane that can swing 360° about its axis.

Disadvantages of slewing cranes include:

(1) high cost, high weight and complex construction;
(2) limited hoisting height and crane boom reach;
(3) higher operator skill required;
(4) higher expenditures for operation and repairs.

The load-lifting capacity of cranes ranges from 1 to 5 tons, but may be higher in some cases. Cranes are installed in pairs at the cargo hatch they serve or along the ship's centre line. Cranes arranged at one side can perform cargo operations only on that side; cranes installed along the centre line can carry out operations on both sides. The crane mechanism is designed so that the load may be hoisted and the crane be slewed to the required angle simultaneously.

Shipboard cranes are usually powered by electric motors; steam and hydraulic cranes are less frequently employed.

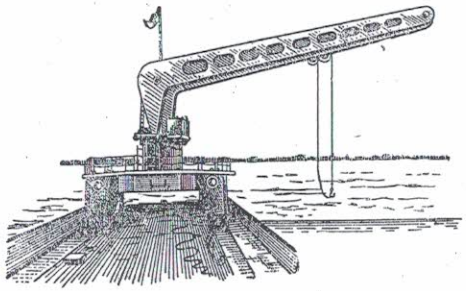


Fig. 180.

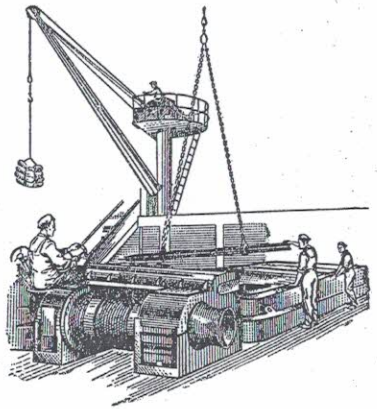


Fig. 181.

Since the engine of a steam crane must slew together with the crane, difficulty arises in supplying steam to the engine. This circumstance, and the higher economy of electric cranes have led to their almost exclusive application on new ships.

The following types of cranes are used on ships:

(1) balance crane (Fig. 177);
(2) fixed pillar crane (Fig. 178);
(3) pivot slewing crane (Fig. 179);
(4) travelling crane (Fig. 180);
(5) above-deck crane (Fig. 181).

Balance cranes are fully counter-balanced but are considerably heavier than other types. Such cranes can travel on a truck along track laid on the deck and are therefore particularly suitable for various transshipping operations as are travelling cranes which are just as heavy as the balance type.

Notwithstanding the lack of balance of pillar type cranes, they find wide application on shipboard because of their light weight and simpler construction. Cranes slewing on a fixed pillar are of lighter weight than the pivot slewing type. The load lifting, boom luffing and crane slewing mechanisms are furnished with brakes. An additional brake is provided on the slewing gear to prevent swinging of the crane under the action of external forces when it is not in operation. This same mechanism is provided with friction clutches enabling the crane to swing about its axis from heavy jerks without damaging the locked gearing of the slewing mechanism.

The energy consumption in crane operation ranges from 0.07 to 0.347 kWh per ton of cargo handled and depends upon the scale of cargo operations.

6-2. Towing Winches

Towing winches are installed in the stern of transport ships and amidships on tugs. A towing winch must be designed to perform towing operations both in calm and heavy seas. Towing in rough weather requires the use of special devices to absorb the dynamic loads that develop in the tow rope. The spring-type buffers used in winches for river craft, which relieve sudden loads by paying out the rope over a fraction of a turn of the winch barrel, prove unsuitable on sea-going vessels.

Soviet practice in the maritime towing of floating docks and nonpropelled vessels has established that several full turns of the winch barrel are required to properly counter-balance the forces acting on the tow rope in heavy seas.

Towing winches are equipped with an automatic device which pays out the rope upon increase in its load and heaves it in when the load is reduced. This device brings the forces acting on the ends of the rope into equilibrium.

The heaving-in speed of automatic marine winches varies from 0.33 to 0.5 m per sec.

Fig. 182 illustrates a steam towing winch with a device for automatically regulating the length of the tow rope in accordance with the variations in the forces applied to the rope.

Through the double spur gearing consisting of gears *13*, *14*, *15* and *16*, a two-cylinder steam engine with cylinders *2* drives the load barrel *1*. Gears *13*, *14* and *16* are keyed on their shafts while gear *15* is mounted freely on its shaft *9*. This gear is engaged with its shaft by claw clutch member *10* which slides along a key on shaft *9*.

Warp ends, or winch heads, mounted in separate supports secured to the deck, are engaged on both ends with shaft *9* by clutch members *12* (only the clutch members secured to the shaft are shown). This arrangement brings the warp ends nearer to the ship's side for convenience in warping operations. Moreover, short lengths of shaft are easier to manufacture, repair, erect and replace than longer ones.

During towing operations clutch member *10* is engaged with gear *15*. In warping, when the winch barrel *1* should be stationary, clutch member *10* is disengaged from gear *15* and shaft *9* is driven from the engine through gears *13* and *14*.

A steady force is established on the tow rope during towing operations in calm weather and the barrel is locked by a band brake applied through crank handle *8*. In this case the steam supply is cut off from the engine.

The brake band fits in a groove of rim *11* on the barrel. The brake has buffer springs *28* and its design differs in no way from that of ordinary band brakes employed on windlasses and winches.

In rough weather towing, the brake is released and the live steam line is opened.

Thus, in this case, the resistance of the vessels being towed is transmitted through the rope to the winch barrel and further, through the gearing and crank gear of the steam engine to the piston. This force, acting on the piston in one direction, is counter-balanced by the pressure of the steam acting in the opposite direction, thereby holding the barrel stationary.

In rough weather, the resistance to movement of the towed vessel will vary due to the dynamic loads resulting from wind and wave action. These dynamic loads on the rope are absorbed by the action of the automatic device *5*.

Live steam is supplied to the engine through pipeline *17* and is admitted to the starting valve when the automatic device is turned off. This valve is controlled with lever *7* and permits the winch to be reversed.

When the automatic device is turned on, steam is admitted through the valve with handwheel *6* into chest *5* of the device and then passes to the starting valve.

The automatic device operates as follows. Under the action of the increased dynamic load the rope turns the winch barrel and, with it, shaft *29* and bevel gear *30* which meshes with gear *33*. Bevel gear *33* is freely mounted on shaft *31* and can be engaged with it by shifting clutch member *32* along a key on the same shaft using lever *25*.

Gear *33* and clutch member *32*, enclosed in housing *26*, are to be engaged when the automatic device is turned on. In this case, the rotation of gear *30* is transmitted to shaft *31* and handwheel *18* which is keyed on this shaft.

As shaft *31* rotates, cylinder *27* will move axially along the threaded part of the shaft shifting stem *34* of piston valve *35* and the valve itself in the same direction.

According to the position shown in the illustration the valve will move to the left increasing the amount of live steam admitted through flanges *36* and *37* to the starting valve and, consequently, through the valve to the distribution valves of the steam engine.

As soon as the dynamic load on the rope is relieved and drops to the normal value, the engine starts operation and rotates the winch barrel in the corresponding direction. Shaft *29* will turn in the same direction actuating the automatic device in winch valve *35* is returned to its initial position when the tow rope is heaved in to its normal length.

The pressure of the steam admitted into the engine cylinders counter-balances the resistance to movement of the towed vessel until the next dynamic load appears, after which the automatic device begins operating again.

After disengaging clutch member *32* from gear *33*, the engine can be started by turning handwheel *18* in either direction depending upon the position of the starting valve.

Movable idlers are provided to uniformly wind the rope on barrel *1* and to prevent adjacent turns of the same layer from being wound on each other.

During winch operation, the idler carriage *3* travels back and forth along rods *19* which are parallel to the winch barrel axis. Tow rope *20* runs between the rollers *21* and *22* and is wound on the proper place on the barrel as guided by these rollers.

The idler carriage is traversed by the action of screw *23* on the carriage drive dog *38* which operates as a nut mating with the screw. Screw *23* is driven from the barrel through a transmission (not shown in the illustration). The thread of screw *23* consists of an endless right- and left-hand groove into which the drive dog *38* fits. The screw rotates constantly in one direction. The carriage is automatically reversed at the ends of its travel when the dog reaches the guide member *24* at each end of the screw which swivels the dog and guides it from the part of the groove with one hand to the part with the opposite hand. Swivel of the dog and transition from the right- to the left-hand parts of the groove and vice versa enable the dog to be used as a nut with right-hand thread when the carriage travels in one direction and as one with left-hand thread after the carriage is reversed.

Power calculations of a towing winch begin by selecting the length and size of the tow rope from Table 60 in accordance with the characteristic of the vessel to be towed, as computed from formula (386). The pulling force developed by the winch barrel should not exceed

$$T_b = \frac{R_{br}}{8}$$

where R_{br} = breaking strength of the tow rope.

The length of the tow rope ranges from 135 to 300 m. The rope capacity of the winch barrel is calculated to accommodate 80 per cent of the total length of the rope.

It should be noted that an increase in the design diameter of the barrel due to the layers of rope wound on it increases the torque on the barrel shaft and the heaving-in speed, while a reduction in the design diameter reduces these values. The number of turns accommodated in one layer on a barrel with a length L_b is

$$m = \frac{L_b}{a_b + d_r}$$

where $a_b \approx 1$ mm = clearance between the turns of rope.

The remaining calculations are done according to equations (415) through (427). If the main characteristics of the steam engine are given, the pulling force developed by the towing winch is determined from formula (432).

The power of the motive unit required to drive a towing winch can also be calculated from the formula

$$N_i = \frac{T_b v_h}{60 \times 75 \eta_w \eta_m} \quad \text{metric hp}$$

where $v_h \approx 0.33$ to 0.5 = heaving-in speed of the rope, m per sec.

Knowing the torque developed on the motive unit shaft and assuming that it runs at constant speed we can plot a curve showing the variation of the pulling force with the design diameter D_{bd} of the barrel using the formula*

$$N_e = \frac{M_m n_m}{716.20} = \frac{M_b n_b i_w}{716.20 i_w \eta_w} = \frac{M_b n_b}{716.20 \eta_w} = \frac{T_b D_{bd} n_b}{2 \times 716.20 \eta_w} \quad \text{metric hp}$$

6-3. Ash Hoisting Winches

Various ash-disposal devices are employed to extract ashes and cinders from the boiler room and discharge them overboard.

Cinders are usually hoisted to the deck with hand, steam or electric ash hoisting winches and are discharged overboard by hand.** A vent duct commonly serves as the hoisting shaft. Steam winches are driven by single-expansion duplex steam engines with a piston stroke and cylinder diameter ranging from 100 to 120 mm and running at 80 to 300 rpm. The engine shaft is usually linked through worm gearing with the winch barrel whose diameter may range from 180 to 300 mm. The ratio of the worm gearing may vary from 10

* This formula is applicable to any type of winch.

** Ash-disposal devices of the ejector type are classified as jet pumps.

to 20 for a rope heaving-in speed of 0.4 to 0.8 m per sec. The engine is reversed by the starting valve. Load lifting capacities of ash hoisting winches range from 100 to 400 kg.

The construction of a steam-and-hand ash hoisting winch is shown in Fig. 183.

Worm *8* is mounted on the shaft *6* of a single-expansion two-cylinder vertical steam engine. This worm meshes with worm wheel *9* mounted on shaft *10* of the winch barrel *7*. The barrel is freely mounted on the shaft together with ring *11* which has a lug *12*. Bolt *13*, located between projections *14* of the barrel, passes through a slot in lug *12*. Ring *11* and ratchet wheel *15* are secured on the shaft by a single key *16*. Pawl *17* of ratchet wheel *15* is pivoted on a pin between lugs *29* of the barrel. This pawl is disengaged from the wheel and held in the disengaged position by hook *18* when the winch is to be driven by the steam engine. Flat spring *30* presses the pawl into engagement with the teeth of the ratchet wheel for the hand drive. Worm *19*, also keyed on shaft *10*, meshes with worm wheel *20* on vertical shaft *21* whose lower end is threaded and carries the nut *22*. Lever *24*, which actuates the differential valve *25*, is mounted on trunnions *28*. The engine is started by shifting the differential valve *25* from its central position, turning lever *24* about trunnions *28*.

The movement of the pistons is transmitted through piston rods *5* and connecting rods *4* to crank disk *26* and further through worm *8*, worm wheel *9*, shaft *10* and ring *11* to the barrel *7*. At the same time, worm *19* rotates vertical shaft *21* through worm wheel *20*.

Axial movement of shaft *21* is restricted so that upon its rotation, nut *22* will travel vertically. Its action, through trunnions *28*, on lever *24* returns the differential valve *25* to the central position. Here we have the same servomotor principle that is employed in steering gear.

If ashes are to be hoisted by hand, bolt *13* with its nut is shifted along the slot of lug *12* towards the centre to bring it out of engagement with the projections *14* of barrel *7*. The pawl *17* is placed into the position shown in the illustration and crank handle *27* is secured to the winch barrel. The winch is driven with this handle.

The same formulas are used in ash hoisting winch design as for other types of winches.

6-4. Boat Winches

Boat-handling gear is used for:

(1) communications between the ship and shore when the ship is riding at anchor in a roadstead;

(2) saving the passengers and crew in case of a shipwreck.

The boat-handling gear must include:

(1) provisioned boats of the proper type, size and quantity as stipulated by the Shipping Register;

(2) a device driven by hand and by machinery powered with motive units for rapidly and safely swinging out and lowering boats to set them afloat and for hoisting them aboard the ship;

(3) gear for stowing the boats in cruising order.

The cubic capacity of the boats depend upon their purposes and is calculated from the expression 0.6 LBH,

where L = maximum length of the boat at the height of the gunwale, m

B = maximum breadth of the boat as measured between the external surfaces, m

H = height of the boat as measured from the garboard strake at the upper false keel to the upper edge of the rubber at the middle of the boat's length, m.

Boats may be made of wood, plastics or metals. The latter are built in the same manner as steel ships or are pressworked as two halves which are then joined along the centre line.

Wooden boats are made of oak, ash, elm, larch, pine or chestnut. Metal and wooden boats are of about the same weight.

A disadvantage of steel boats is their rapid corrosion, of wooden boats—damage by woodworms.

Hand-operated machinery is usually installed to hoist boats weighing up to 510 kg.

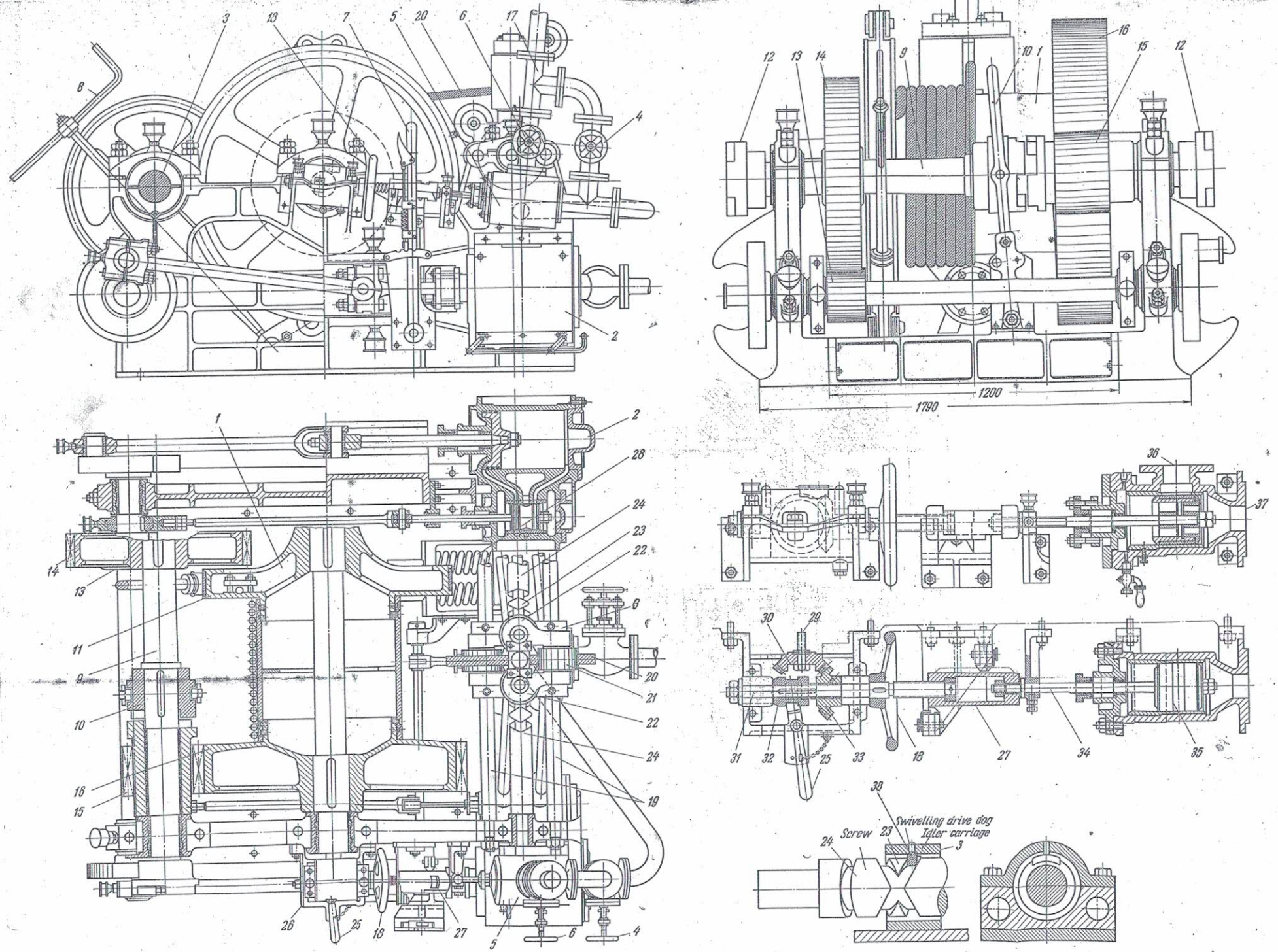
Heavy boats, weighing about 510 kg and more, are hoisted by power-operated winches with nonreversible worm gearing.

Steam and electric boat winches have found the most extensive application. Steam winches are less economical than the electric type due to the steam condensation losses in the steam lines and steam engines. Moreover, it is necessary to warm up the steam engine before using the winch while electric winches are always ready for action. The steam engines of boat winches may be designed to operate either with or without steam expansion. In the former case the steam cut-off value ranges from 0.4 to 0.8.

Fig. 184 illustrates a boat winch powered by electric motor *1* whose shaft is linked to the shaft of worm *3* by the flexible coupling *2*. The half-coupling of the worm shaft has a brake drum on which brake band *4* is fitted. This band is linked through a lever system with the electromagnetic braking device *5*. The worm shaft runs in ball bearings *6* while axial forces are carried by the double ball thrust bearing *7*.

Worm *3* drives the worm wheel *8* on whose shaft identical winch heads *9* are mounted for winding the pendant of the boat's falls.

The required pulling force of the winch heads is the sum of the weights of the boats with rigging and the passengers, taking into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37
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29
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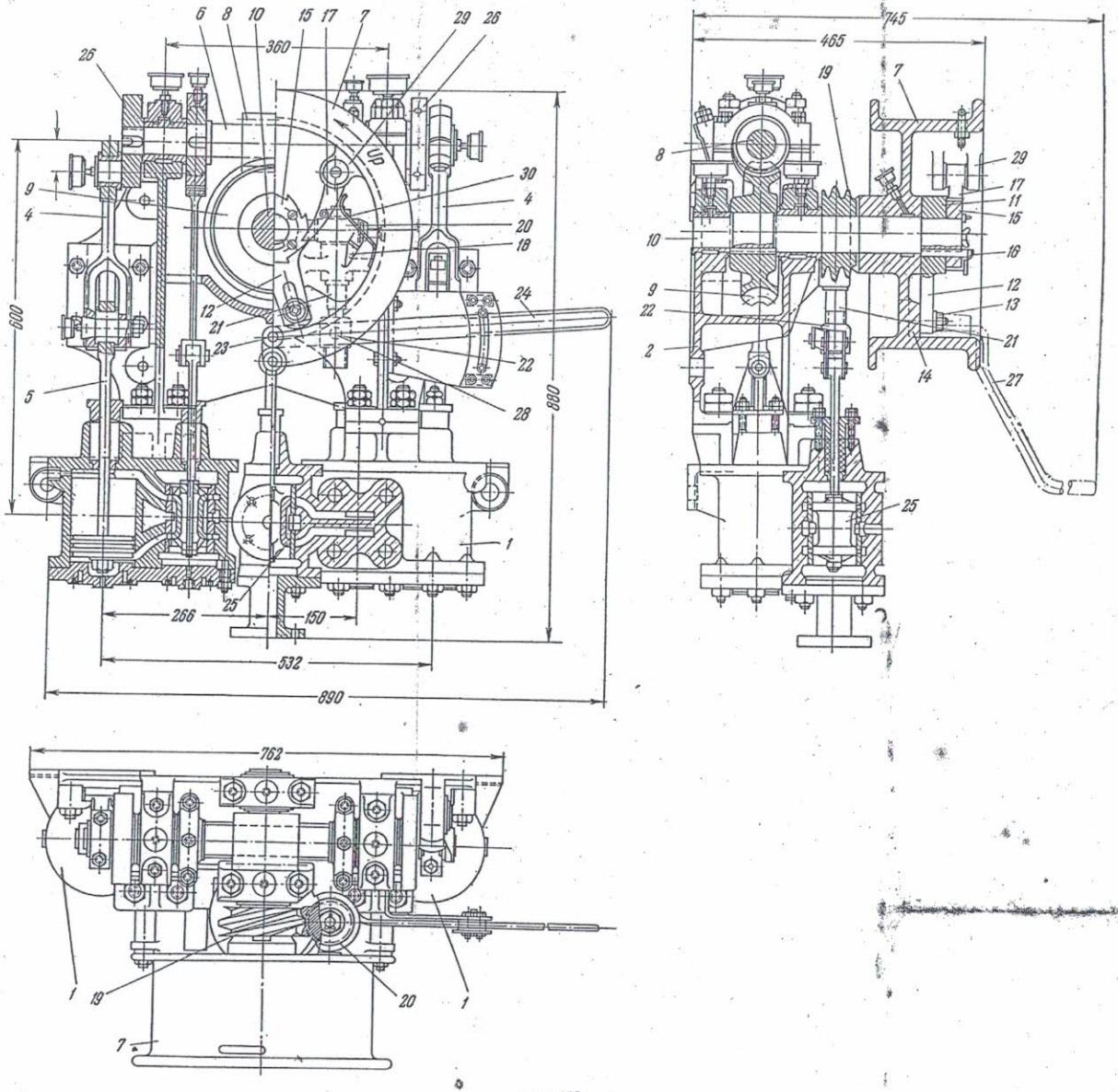
38
Swivelling drive dog
Screw
23
Idler carriage
24
3

Fig. 182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266
150
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2
21
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Fig. 183

consideration all parasitic resistances in the boat's falls and in the various guide blocks through which the tackle fall runs to the winch heads.

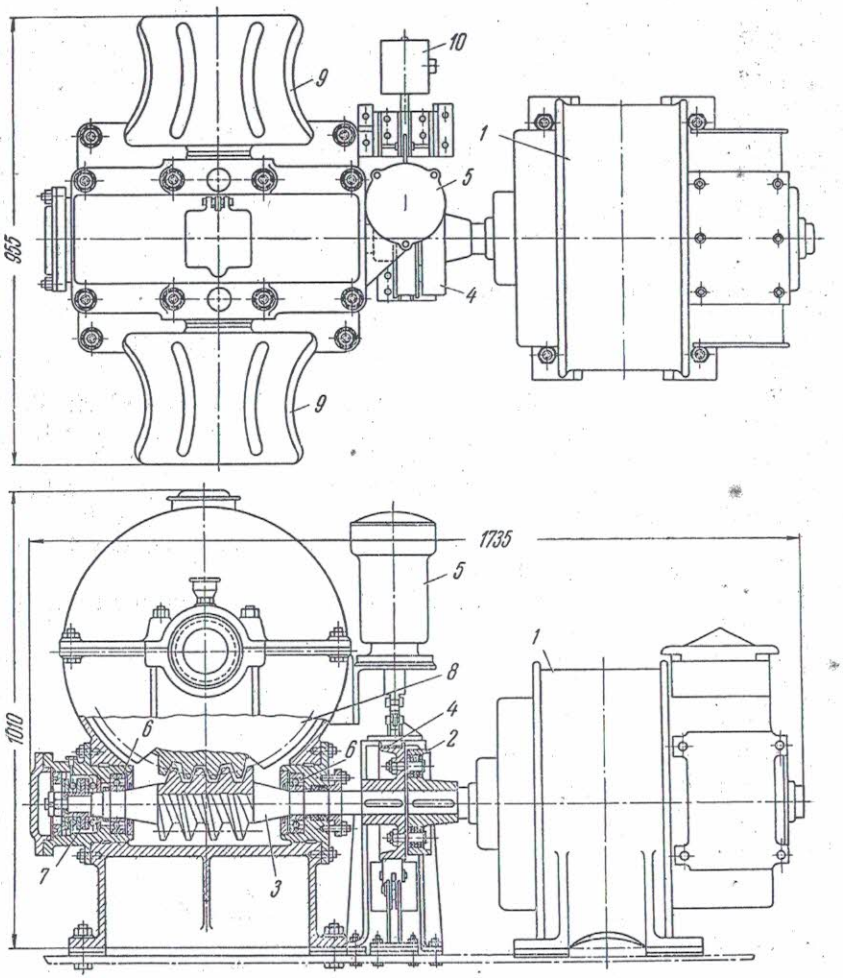


Fig. 184.

The design load Q acting on the falls of one davit is determined from the formula

$$Q = 0.5\,(Q_b + Q_p k_n) + Q_f$$

where Q_b = 570 to 2,175 = weight of the fully rigged boat, kg
Q_p = total weight of all persons allowed to embark (the weight of one person is approximately 75 kg; the number of persons in a boat may reach 78), kg
$Q_f = 0.05(Q_b + Q_p)$ = weight of the boat's falls, kg
k_n = 0.9 to 1.1 = coefficient of nonequal distribution of the movable load due to the weight of the persons in the boat.

The maximum tension of the fall at the winch head, after running over the maximum number of guide devices, is

$$T_{max} = \frac{0.5(Q_b + 1.1Q_p) + Q_f}{m\eta_f \eta_r \eta_s^a}$$

where m = total number of blocks in the boat's falls;

$$\eta_f = \frac{1}{m\varepsilon^m} \frac{\varepsilon^m - 1}{\varepsilon - 1} = \text{efficiency of the boat's falls}$$

ε = coefficient depending upon the ratio of the block diameter to the tackle fall diameter (ε=1.1 for a hemp fall and ε=1.04 to 1.06 for a steel wire rope)
η_r = 0.9 to 0.97 = efficiency of the davit guide roller
η_s = 0.9 to 0.97 = efficiency of the snatch-block
a = maximum number of blocks between the davit guide roller and the winch head.

The tension at the end of a rope that has run over the minimum number of blocks is

$$T_{min} = \frac{0.5(Q_b + 0.9Q_p) + Q_f}{m\eta_f \eta_r \eta_s^c}$$

where c = minimum number of blocks.

The diameter, d_f, of a hemp fall is selected according to the breaking strength $(T_{max} + T_{min})\, 6 \leqslant R_{br}$ as a function of the boat length from Table 63 (U.S.S.R. Shipping Register).

Table 63

Boat length, m	Circumference of tackle fall, mm	Breaking strength, kg
8.25 to 9.14	95	6,100
7.62 to 8.25	89	5,400
7.35 to 7.62	83	4,600
6.72 to 7.35	73	3,900

The winch head diameter is

$$D_h = (5 \text{ to } 8)\, d_f$$

The speed, v_b, with which the boats are hoisted and lowered is assigned so that if the ship is rolling one of these operations can be carried out during the rolling period. This can be accomplished if the heaving-in speed is greater than the vertical component of the crest speed of waves running along the length of the ship. It has been established that the boat hoisting speed must be at least v_b = = 0.15 m per sec under these conditions. The heaving-in speed of the tackle fall when single-sheave blocks are used must in this case be v_f = 0.3 m per sec.

The required winch head speed is found from the equation

$$\pi (D_h + d_f)\, n_h = 60 v_f$$

$$n_h = \frac{60 v_f}{\pi (D_h + d_f)} = 19.1 \frac{v_f}{D_h + d_f} \quad \text{rpm}$$

Assigning a motive unit speed (n_m = 500 to 1,600 rpm for electric motors and n_m = 200 to 380 rpm for steam engines), we can find the gearing ratio of the boat winch. Thus

$$i_{bw} = \frac{n_m}{n_h}$$

In designing nonreversible worm gearing the number of teeth on the worm wheel is taken in the range from 24 to 44. The pulling force on the winch head is equal to the sum of the pulling forces on the tackle falls:

$$T = T_{max} + T_{min}$$

Disregarding friction losses, the torque developed on the winch head shaft will be

$$M_h = \frac{T(D_h + d_f)}{2}$$

If the winch has an efficiency of η_{bw}, the torque and power on the motive unit shaft will be

$$M_{mb} = \frac{M_h}{\eta_{bw} i_{bw}} = \frac{T(D_h + d_f)}{2\eta_{bw} i_{bw}}$$

and

$$N_e = \frac{M_{mb} n_m}{716.20} \quad \text{metric hp}$$

The cylinder diameter and indicated power of steam boat winches are determined from the same Posdyunin formulas used in

determining these values for other steam-powered auxiliary machinery.

The required motor power rating for an electric winch can be calculated from the formula

$$N_e = \frac{(T_{max} + T_{min})\, v_b}{75 \times 60 \eta_{bw}} \quad \text{metric hp}$$

6-5. Winch Operation

Before starting a winch it is necessary to make sure by inspection that it is in order and that no foreign objects hinder the moving parts of the winch. Then, warm up the steam supply line and the winch engine, apply lubricant to all friction surfaces, check the starting gear by turning it to the "hoist" and "lower" positions, check whether the speed-changing clutches engage and disengage properly and whether the braking gear is in order. After this, test the forward and reverse operation of the winch; if no knocking is heard and reversal is rapid and smooth, the winch is ready for regular operation. The winch is started and stopped and the shaft speed is changed in operation by opening and closing a stop valve or the starting valve. The winch is reversed either by shifting the links of the reversing gear or by operating the starting valve.

During winch operation it is necessary to see that lubricant is being properly fed, to check the temperature of the parts subjected to friction and to listen for knocking. As soon as abnormal noise is heard, stop the winch, find and eliminate the cause of the noise.

The load lowering speed should be regulated by applying the brake. Backsteam should be resorted to only when the load drops too fast even after applying the brake.

If hoisting operations are interrupted for short periods in winter, the winch should be run idle at low speed with open blow-off valves.

If the winch is not to be operated for a prolonged period in winter, it is necessary to drain the condensate from the cylinders and the live and exhaust steam lines.

Winch operation is prohibited if cracks are found in critical parts, if the motive mechanism, steam distribution or breaking gear is out of order or if the gears are excessively worn or some of the teeth are missing.

Part Three

SHIPBOARD SYSTEMS

Chapter 7

GENERAL SYSTEMS

7-1. Classification and Functions of Shipboard Systems

Any shipboard system comprises a network of special pipelines with their component machinery, apparatuses and instruments, and is designed to perform definite functions on the ship. Shipboard systems may be classified as general, special and marine power plant systems.

The classification and functions of general shipboard systems and the special systems of tankers and ice-breakers are indicated in Table 64.

Pipeline elements ensure airtight joints between the separate components and parts of a pipeline system and otherwise enable the system to serve the purpose it is designed for.

Elements of pipelines include:

(a) piping, which is the main element of a line and connects the point where a fluid is received to the point of discharge;

(b) connecting, or line, fittings which connect the separate pipes and components of a pipeline together and to the ship's hull (flanges, tees, elbows, bulkhead castings, deck plates, and couplings);

(c) disconnecting and regulating fittings (valves) which serve to connect, disconnect or switch over either separate sections of a pipeline or the whole pipeline, and also for regulating the pressure and controlling the flow in a pipeline;

(d) valve gear which provides for control of valves and other fittings either from their place of installation (local control) or from a distance (remote control);

(e) other fittings which find application in certain systems (pipeline inlets, mud boxes, inlet screens, separators, steam traps, drenchers, sprinklers, etc.).

The main features that characterize piping and determine its purpose are: inside diameter, wall thickness and the material of which the pipe is made.

With a given length and rate of flow of liquid through a pipeline the velocity of flow in the given system and, consequently, the loss of head along the length of the line are determined by the inside diameter of the piping.

The wall thickness and material of the pipe determine its strength and, therefore, the permissible pressure in the pipeline. The corrosion resistance of the pipe also depends upon the material. Pipes supplied for ship-building are from 4 to 7 m long and may be either rigid or flexible. Steel, copper and brass pipes are used for marine installations. Flexible metal piping and piping of rubberized or specially impregnated fabric with fillers imparting rigidity are finding wide applications for the main pipelines. This piping is much simpler to install. Three types of pressure are specified for piping: test, conventional and working pressures.

The test pressure is the maximum momentary pressure that the pipe must withstand in a hydraulic water test at a definite temperature.

Conventional pressure is the pressure of the fluid at normal temperature which corresponds, insofar as the pipe strength is concerned, to the pressure of the fluid at some higher temperature. The conventional pressure takes into consideration the reduction in the mechanical properties of the pipe material and that of the connecting elements due to the increased temperature of the fluid flowing through them.

The working pressure is the maximum permissible pressure in the pipe in actual operating conditions of the given system.

Since the diameters of flanges and fittings depend upon the outside diameter of the pipe, the concept of nominal diameter has been introduced to facilitate the standardization and selection of the components of a pipeline. Each nominal diameter corresponds to one constant outside diameter while the inside diameter depends upon the thickness of the pipe walls.

Pipe joints may be either permanent or separable. The former are made of riveting, welding or brazing, the latter are flange, union or coupling joints. The two last types are employed for pipe of small size. Flanges are attached to the pipes by threads, expanding, welding or brazing.

Klingerite, paranite, asbestos, fibres, rubbers, pressboard and soft metals are used for jointing material between the flanges.

Table 64

Name			Functions
General shipboard systems	Bilge systems	Ballast	Maintaining normal draught and trim of the ship's hull, and eliminating heeling by filling or emptying the ballast tanks or transferring the water ballast from tank to tank in ballasting the ship.
		Drainage	Removing water and liquid residues from the compartments of the hull.
	Sanitary systems	Salt (sea) water	Delivering salt (sea) water to its destination (W. C.'s, bathrooms, showers, etc.).
		Fresh washing water	Supplying water from the standby tanks to its destination (bathrooms, showers, baths, galley, lavatories, laundries, refreshment bars, etc.).
		Fresh drinking water	Supplying water from the standby drinking water tanks to its destination (galleys, refreshment bars, staterooms, drinking fountains, etc.).
		Sewage	Draining fecal liquids and sewage from urinals and latrines.
		Sanitary	Draining sewage from the baths, laundries, lavatories, showers, victuals stores, etc., to the sanitary tanks, or wherever it is accumulated.
		Ventilation	Maintaining air of the required quality in the spaces and holds of the ship.
		Heating	Providing and maintaining the required temperature in the spaces during cold weather.
		Fire	Fighting fires.
		Scupper	Discharging water overboard from the upper decks.
		Water discharge (emergency)	Discharging large amounts of water flooding the hull as the result of an accident.
Special systems of icebreakers		Trimming	Facilitating the elevation of the ship's bow onto the ice by producing a trim by the stern. This is done by taking in liquid into the aft trimming tanks after the ice-breaker climbs onto the ice, thereby increasing the weight of the bow so as to break the ice.
		Heeling	Freeing the hull of the ship when it becomes icebound by heeling from side to side.
Special systems of tankers		Cargo	Loading and discharging liquid cargo and transferring it from tank to tank.
		Evacuating	Removing cargo residues that cannot be handled by the cargo system since its inlet connections are 100 to 150 mm above the tank bottom.
		Cargo heating	Heating thickened viscous liquid cargo to a state where it can be handled by the pumps.
		Gas discharge	Maintaining the required pressure in the tanks by discharging vapours that are evolved when the temperature of the cargo is raised in warm weather; evacuating the air when cargo is being taken into the tanks and preventing a vacuum from being created when the cargo is discharged from the tanks.
		Deck flooding	Reducing the loss of liquid cargo through evaporation by flooding the deck with sea water to lower its temperature.

Table 65

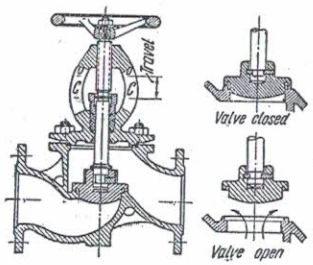
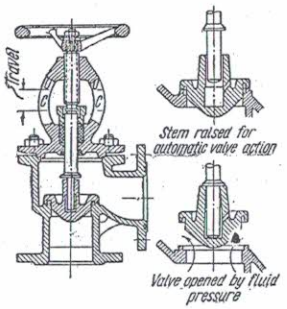
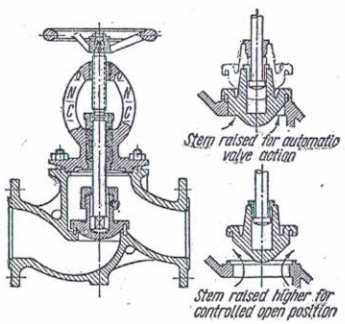
Type	Construction	Features
Stop valve		The valve disk is linked rigidly to the stem and can be actuated only from the handwheel. Serves as a shutoff device.
Screw-down nonreturn (check) valve		The valve disk is not fixed to the stem and can slide along it. When the stem is screwed down tight the valve acts as a shutoff device. When the stem is raised the valve operates as an adjustable-lift check, or nonreturn, valve. The stem does not raise the disk.
Controlled nonreturn valve		The valve disk is freely mounted on the stem but its lift, without raising the stem, is limited. The free travel of the disk along the stem enables the valve to operate as an adjustable-lift check valve. The limited nature of the travel allows the stem to lift the disk whenever required.

Table 65 (continued)

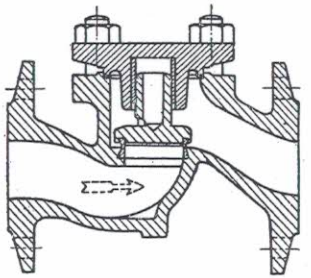
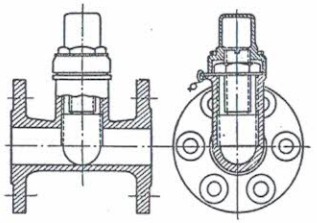
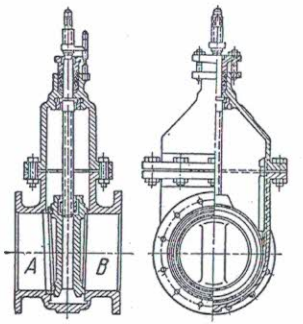
Type	Construction	Features
Nonreturn (lift-check) valve		Allows liquid to flow only in one direction. The valve disk is lifted from or returned to its seat by the pressure of the liquid acting on the corresponding surface of the disk.
Throttle valve		Serves to reduce the pressure of the liquid by increasing its velocity and reducing the passage area. It does not ensure constant pressure after the valve in cases when the pressure before it is variable.
Gate valve		Serves as a shutoff device for large size pipelines. It can be used as an overflow valve for passing liquid from one compartment to another.

Table 65 (continued)

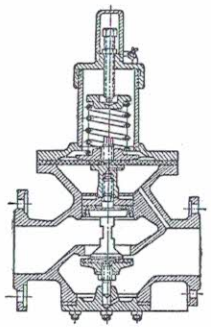
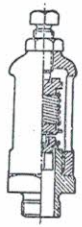
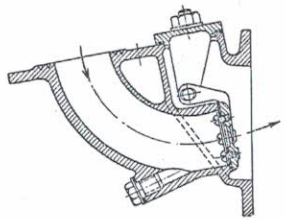
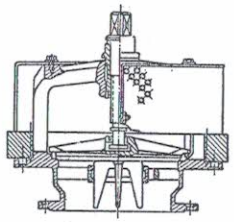
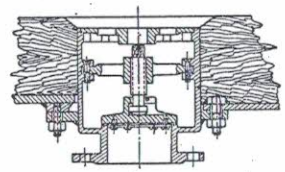
Type	Construction	Features
Reducing valve		Serves to reduce the pressure of the liquid flowing beyond the valve, and to maintain this pressure automatically, but it cannot be used as a shutoff device.
Relief valve		Serves to prevent the pressure in the object on which it is installed from increasing above the permissible value.
Swing-check valve		Used as an automatic shutoff device operated by the pressure of the liquid or for mounting at the end of pipelines discharging at the shipside.
Drain valve		Designed for draining liquid from a space which does not have its own drainage facilities.

Table 65 (continued)

Type	Construction	Features
Scupper valve		Serves to dispose of water from decks located below the waterline.

At places where piping is laid through waterproof bulkheads, decks or flooring, the latter are maintained waterproof by the use of cast or welded bulkhead pieces or deck plates. These pieces or plates may be straight, angle, screwed or flanged. Water and gas pipe up to 80 mm in diameter, operating at a fluid pressure up to 16 kg per sq cm, has fittings, elbows, tees, couplings and other accessories covered by the pertinent U.S.S.R. Standards. Pipes larger than 80 mm in diameter are joined by fittings of welded design. The direction of a pipeline is changed by bending the pipes.

Expansion joints are provided in pipelines to reduce the stresses in the pipe due to thermal expansion or deformation of the ship's hull. The classification, functions and construction of disconnecting and regulating fittings and the most widely applied types of remote-control gear are given in Fig. 185 and Tables 65 and 66.

Table 66

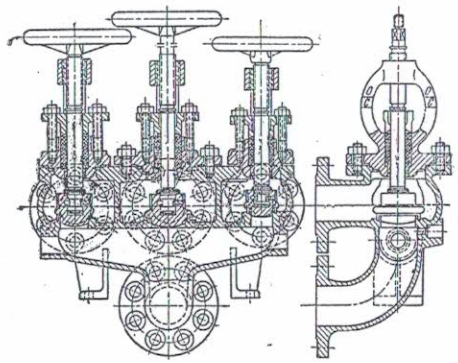
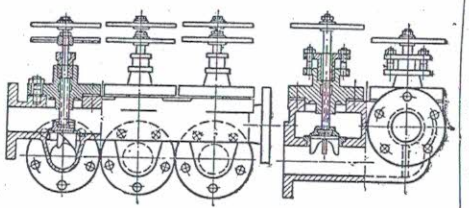
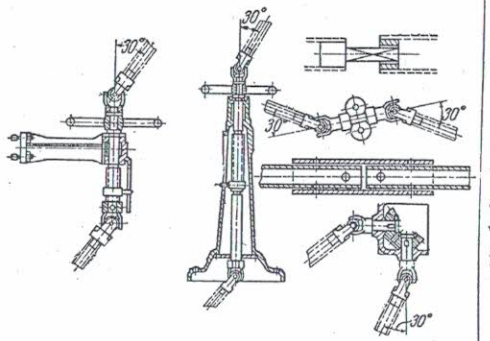
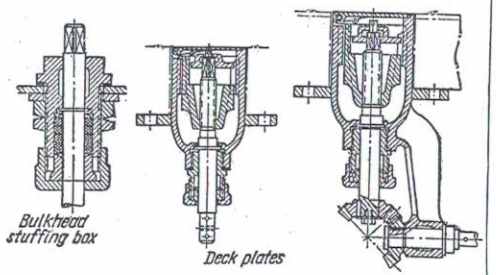
Construction	Type and features
	Valve chest with common liquid inlet and separate outlets. This type of chest with a common inlet to all the valves and a separate outlet from each valve is usually mounted in the discharge line of a pump serving several objects and connects the object being served at the given time to the pump.

Table 66 (continued)

Construction	Type and features
	Valve chest with separate inlets and a common outlet. This chest with a separate inlet to each valve and a common outlet from all the valves is usually mounted in the suction line of a pump serving several objects and connects the object being served at the given time to the pump.
	Rod-type remote-control gear. Remote-control gear is designed for convenience in controlling fittings located in places difficult of access, and also in emergencies when access to the local gear is impossible.
	Bulkhead stuffing boxes and deck plates. These fittings are used to ensure the watertightness of bulkheads and the deck at places where shafts or rods pass through them.

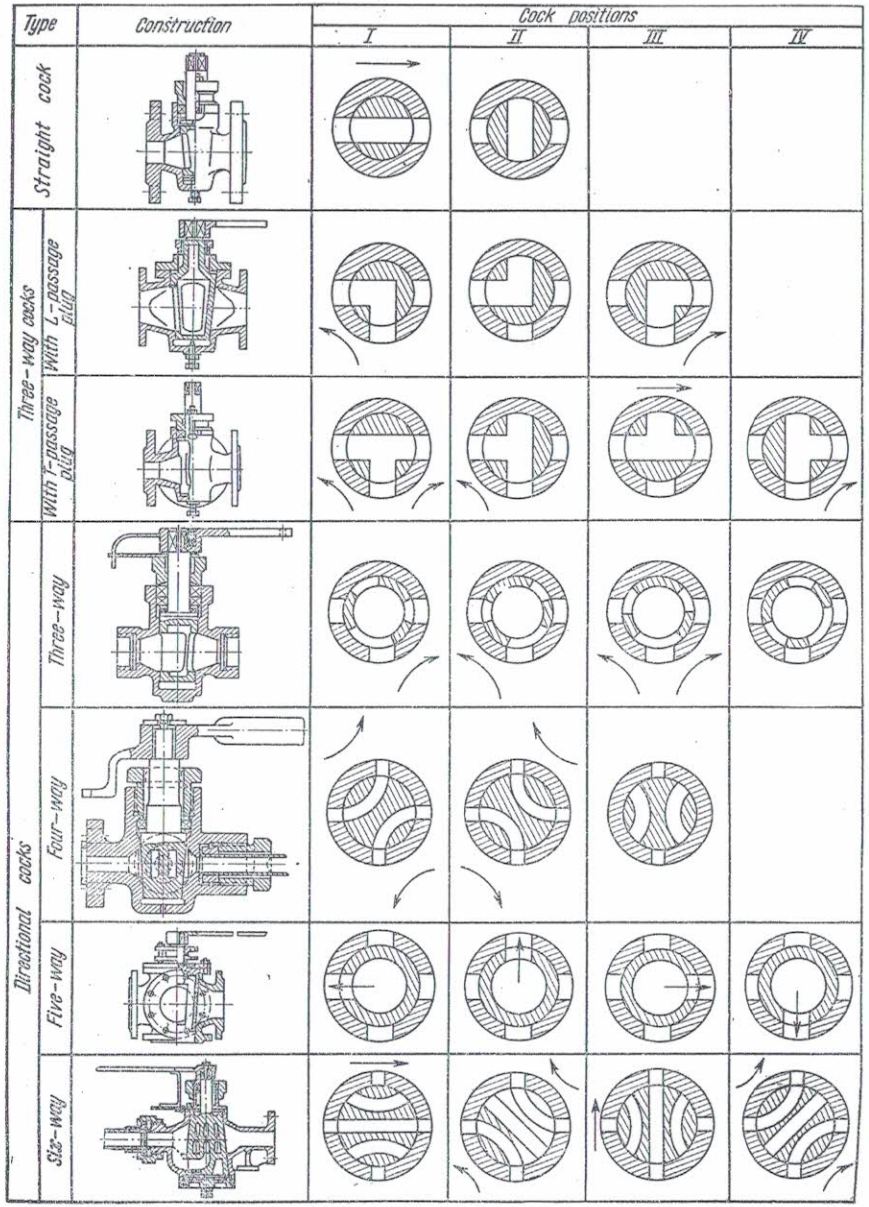


Fig. 185.

7-2. General Requirements for Pipelines of Shipboard Systems

The U.S.S.R. Shipping Register stipulates that all pipelines of the general shipboard systems must meet the following requirements:

1. All pipelines shall be provided with supports, not interfering with the thermal expansion of the piping, to prevent shifting and sagging of the pipes.

2. All pipelines shall be provided with devices for relieving the stresses due to their thermal expansion. If these devices are simply expansion bends, the distance between the centres of the bends shall be at least three pipe diameters and the length of the bent part shall be at least eight pipe diameters.

3. At places where they pass through watertight bulkheads, the pipes of all shipboard systems shall be fastened to the bulkheads by means of bulkhead pieces, or welded or riveted flanges. Fastening the pipe flanges to the bulkheads with through bolts shall not be permitted.

4. All piping passing through cargo holds, coal bunkers, chain lockers and other spaces (except the machine boiler rooms) shall be protected against damage by casings.

5. As a rule, piping shall not pass through oil fuel tanks. However, if this cannot be avoided, a tunnel shall be provided, unless the pipes are tested with hydraulic pressure after each two years of operation.

6. Shipside fittings, gate valves, cocks and other valves for various purposes shall be designed so as to prevent accidental penetration of sea water into the ship and shall be arranged wherever possible above the access boards of the machine and boiler rooms.

7. Shipside fittings located below the load waterline and having detachable handles shall be designed so that the handle can be removed when the valve is closed.

8. All disconnecting fittings shall be designed so that one can see at a glance whether they are open or closed.

9. Piping and fittings shall not be fastened directly to watertight bulkheads; they may be fastened to oiltight bulkheads if this is done before the tanks have been tested.

10. All sewage outlets shall be, as far as possible, on the same, preferably the port, side of the ship. It is not advisable to arrange the pump inlets on the same side.

11. The outlet openings shall be aft of the sea water inlets along the ship's side.

12. All shipside inlet openings shall be covered with an external grid or screen to prevent debris being taken in and choking the passages of the bottom valves.

13. Bottom inlet fittings shall be equipped with a steam or air blow-off system having a pressure not exceeding 3 kg per sq cm. All fittings and outlet openings that may be frozen over shall be equipped with a steam heating system.

7-3. Bilge Systems

(A) DRAINAGE SYSTEM

The function of the drainage system is to remove small amounts of water that collect in the hull and are concentrated in the bilge courses and wells. The sources of this water are sweating of the plating, infiltration through impaired seams of the hull, and penetration of water through openings in the deck and freeboard, that are not fully watertight, during rough weather and rain or in hosing the deck or washing the superstructures. Water also enters the bilge courses and wells of the engine and boiler rooms and the propeller shaft tunnels from leaks in joints of the piping and parts of the machinery, from systems for blowing off various machinery, from systems for cooling friction surfaces, etc.

Timely removal of the water that collects in the bilge courses and wells of the cargo holds prevents dampening or wetting of the cargo and reduces the intensity of corrosive attack on the hull and framing. Excessive accumulation of bilge water in refrigerator ships may lead to the deterioration of the insulation in the refrigerated holds.

If bilge water is not regularly removed from the engine and boiler rooms it will reach a level where it impairs normal working conditions of the personnel and the main and auxiliary machinery and boilers. The drainage system is frequently used to remove water from flooded spaces after the leaks are eliminated if more effective means of draining these spaces, for example, special emergency pumps, are unavailable.

The drainage system is to be designed so that any hold and engine and boiler room can be served by any drainage pump. Its arrangement must exclude the possibility of sea water and water from various tanks getting into the cargo holds or engine and boiler rooms or from one space to another. This condition is met by the provision of disconnecting fittings of the screw-down nonreturn type. It is advisable to equip the inlet connections with nonreturn valves. The design of the drainage pipeline should provide for pumping the water directly out of the engine and boiler room with one drainage pump while the remaining pumps are employed exclusively for draining the other compartments and spaces of the ship. Provision must be made, moreover, for disconnecting the drainage pipe-

line from spaces that can be used for ballasting purposes. As a rule, the drainage pipeline should be laid outside of the double-bottom space.

If, however, the drainage pipeline passes through the double-bottom space or a tank, it is to be tested at the same hydraulic pressure as the space through which it passes. The inlets of the drainage pipeline in any given space or compartment should be arranged so as to ensure its complete drainage both when the ship is in the upright position or lists 5° to either side. These inlets are usually installed in the breast bilge courses. In addition to the inlets in the bilge courses, ships with a double bottom, having a slope along the centre line in each space and compartment, should also have an inlet installed in the bilge well at the centre line. Wells should not be excessively deep in a double bottom. In any case, they should be spaced at least 460 mm from the shell plating in passenger ships. If the double-bottom plating extends to the shell plating, the wells are arranged in the breast appendages of the hull. They are two or three frame spaces long and formed by watertight floors and the side stringer.

Screens or grates for water drainage are installed in the double-bottom plating directly over the wells.

The following requirements are made to the drainage of engine and boiler rooms:

(a) In cases when the engine and boiler room is a single compartment without a double bottom and the bottom rises towards the sides at an angle of at least 5°, such a compartment may have one inlet from the drainage system line and one directly leading to the drainage pump. If the rise is less than 5°, additional inlets from the line should be installed at the ship's sides.

(b) In cases when the engine and boiler room is a single compartment and has a double bottom over its whole extent, forming breast bilge courses, it is necessary to provide two inlets at each side served by two mechanically driven drainage pumps. One of the inlets should be connected directly to a drainage pump which is independent of the main engine. If the double-bottom plating extends from side to side the engine and boiler room must be drained by four inlet branches, one being connected directly to a drainage pump and the other three to the drainage line.

(c) In cases when the engine and boiler rooms are in separate compartments, the number and arrangement of the drainage line inlets must be the same as for draining the holds. In the engine room there should be one additional inlet branch connected directly to a drainage pump.

Water is removed directly from the engine rooms of steamships during emergencies through inlet connections to the main circulating pumps. These connections should have nonreturn valves and be capable of draining water from the engine room. The diameter of the connection should be at least two-thirds that of the suction pipes of the circulating pumps.

The suction of the inlet branches of the drainage pipeline should be equipped with inlet screens having holes not over 10 mm in diameter, the total area of the holes being at least thrice that of the inlet pipe cross section. In addition to screens, mud boxes are provided in the drainage pipeline to prevent foreign articles from getting into the pumps together with the water from the bilge courses.

If the forepeak and afterpeak do not serve as ballast tanks, they can be drained either by inlet branches of the drainage pipeline or by means of hand and portable water-jet pumps. The actuating fluid of the latter is supplied from the fire mains.

The drainage system, when used to drain the forepeak through inlet branches, is usually laid outside of the double bottom.

Gate valves, mounted on these branches in the forepeak, are operated from a control station located above the load waterline.

Propeller shaft tunnels are also drained by branches of the drainage system.

(B) BALLAST SYSTEM

Depending upon its purpose ballast is carried in the fore- and afterpeaks, double-bottom tanks, deep-tanks and side tanks.

Ballast taken into the fore- and afterpeaks serves to change the trim of the ship. The double-bottom ballast tanks and the deep-tanks are pumped up with ballast to obtain the proper draught of the hull and to eliminate the list. The side ballast tanks serve chiefly to eliminate the list.

The ballast system is designed on the basis of centralized operation. The ballast tanks are filled and emptied by the same pipeline so that stop valves must be installed in the system. The application of the centralized principle enables the ballast tanks to be filled and emptied and the ballast to be transferred from tank to tank.

By weight, the water ballast ranges from 10 to 17 per cent of the ship's displacement.

The U.S.S.R. Shipping Register stipulates that the ballast pipeline shall be arranged so as to exclude the possibility of sea water or water from the ballast tanks getting into the cargo holds and the engine and boiler rooms. This is accomplished by installing stop and screw-down nonreturn valves.

If compartments of the ship, designed for carrying dry or liquid cargo, are to be temporarily used as ballast tanks, provision must be made for reliably disconnecting the ballast pipeline from these

compartments. The ballast pipeline must not be laid through oil tanks located in the double-bottom space. In the vicinity of double-bottom oil tanks the ballast pipeline is to be laid above the double bottom.

However, if it is impossible to protect the ballast pipeline against damage during cargo handling and carriage, as well as against freezing, it may be laid through the oil tanks of the double bottom, provided the following conditions are complied with: (1) the tanks are equipped with heating facilities; (2) oil-resistant gaskets are inserted between the flanges and (3) the pipeline is tested with a hydraulic pressure of 4 kg per sq cm (such a test is repeated every two years). The ballast pipeline is laid up to the fore- and afterpeak in the ordinary manner. If it runs to the forepeak above the double-bottom space, it must be equipped with a valve mounted on the forepeak bulkhead and operated from a control station located on the bulkhead deck or above it.

The ballast system of tankers must be separated from the cargo system. Pipelines of the ballast system must not be laid through the cargo tanks and pipelines of the cargo system must not be laid through the ballast tanks. The size of the inlet piping of the ballast system should suit the tank of maximum capacity and is selected from Table 3 (see Part One).

Cast iron or welded steel inlet branches in the ballast tanks are mounted on the bottom on lugs arranged around the perimeter of the suction hole in the bell mouth. The bell mouth is installed to bring the face of the inlet nearer to the tank bottom and to reduce the velocity of liquid flow to the inlet. The minimum height of the inlet plane of the bell mouth above the tank bottom and the diameter of the bell mouth are determined by the equality of the area of the cylindrical suction gap and the interior cross-sectional area of the inlet pipe.

Sea water is pumped into the ballast system through a kingston valve installed in the hull bottom or above the breast in the vicinity of the engine room. Ships that navigate in shallow waters or enter rivers are additionally equipped with an upper side kingston valve. Its diameter should be at least two-thirds of the diameter of the main kingston valve while its inlet opening should be located about 300 mm lower than the light line. The opening is usually rectangular in shape and is protected by a grate with rectangular slots from 12 to 15 mm in width arranged in the longitudinal direction in reference to the ship. The total area of the slots in the grate should be from 1.25 to 2 times greater than the clear opening of the pipe. A steam pipeline with a valve having an inner diameter of about 19 mm is laid to the kingston valves to clean the grates and to melt ice that may form on them. Ballast tanks located

below the waterline may be filled either by a ballast pump or by gravity. The overboard discharge outlet of the ballast pipeline is located about 300 mm above the Plimsoll disk and is provided with a swing-check valve.

The ballast pipeline is tested in assembly with its fittings at a hydraulic pressure of 4 kg per sq cm.

Air pockets in filling the tanks and vacuum in pumping out the liquid are prevented by the provision of vent pipes. The latter are best arranged as indicated in Table 67.

Table 67

Type of ballast space	Number and arrangement of vent pipes
Large compartments with a flat ceiling	Four pipes, one in each corner of the compartment
Compartments with a convex ceiling	Two pipes at the ends of the compartment, at the highest point in the ceiling
Small compartments with a flat ceiling	Two pipes arranged crosswise in the port and starboard corners, fore and aft

The ballast tanks have plugs mounted in the shell plating of the hull for complete drainage in dry-docking the ship.

A diagram of the arrangement of the ballast, drainage and salt water pipelines in a sea-going vessel is shown in Fig. 186.

Through inlets *8* the bilge water enters the screw-down nonreturn valve chests *2*. The inlets of the fore compartments are connected to their valve chest *2* which is equipped with remote-control gear *3* brought out on the deck.

Chests *2* are connected by a pipeline to the controlled nonreturn valves *4* and, through them, to the stop valve chests *1* and the bilge and ballast pumps. The discharge connection of the pump is connected to stop valve chest *11* and the nonreturn overboard discharge valve *5*. The engine room is drained through inlet screens *10* and valves *14* of the controlled nonreturn type.

The inlets *9* of the ballast pipeline are connected to stop valve chest *12* and, through gate valve *6* and stop valve chest *1*, to the suction pipe of the pump and the shipside opening. The fore ballast tank has a valve *7*. The ballast tanks are filled through a pipeline connected to the kingston valve through gate valve *13*, stop valve

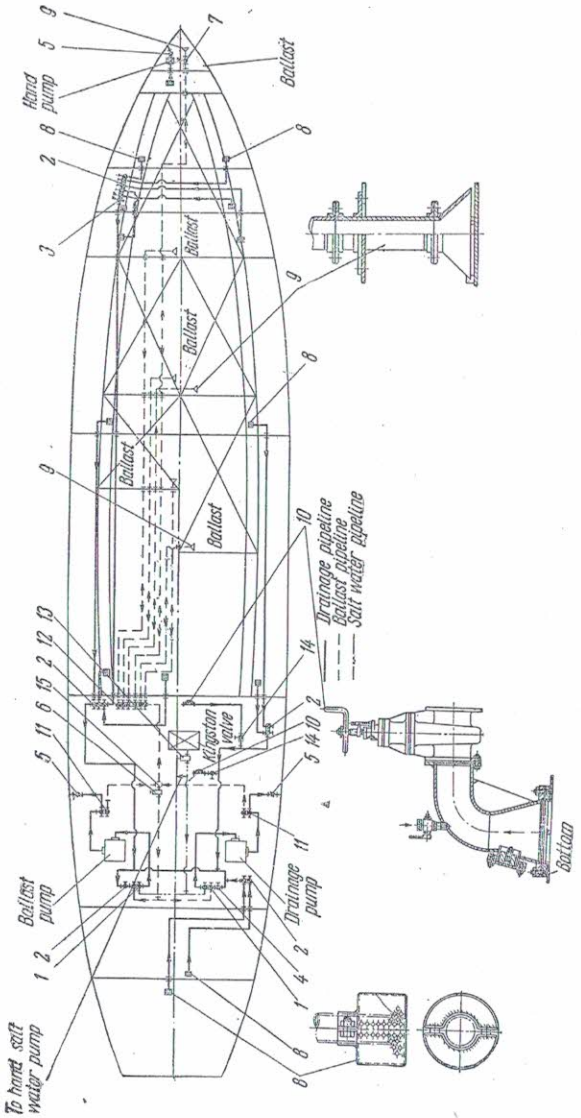


Fig. 186.

chest *1*, the bilge and ballast pump, stop valve chest *11*, four-way valve *15* and valve chest *12* of the ballast pipelines. The forepeak is drained by a hand drainage pump.

7-4. Sanitary Systems

(A) DRINKING WATER SYSTEM

Entirely independent drinking water pipelines are installed to supply water for drinking purposes on a ship.

Drinking water is delivered from the store tanks to the service tanks and from there to the points of disposal. In most cases, the service tanks are installed on open decks.

The service tanks are equipped with heaters and are coated with thermal insulation to prevent freezing of the water.

The store of drinking water is carried in special store tanks remote from sources of heat and systems carrying other liquids to avoid heating and pollution from the penetration of liquids from adjacent

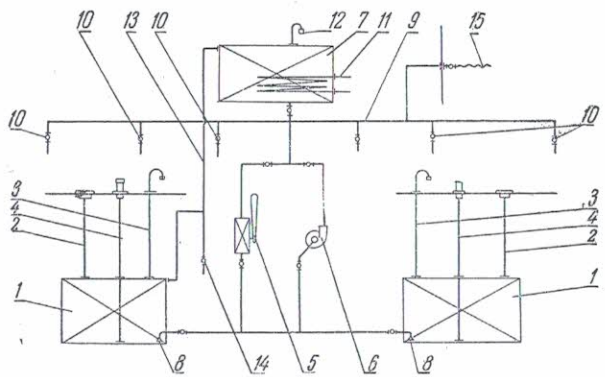


Fig. 187.

tanks. There should be at least two drinking water tanks on the ship. This facilitates maintenance of the proper sanitary conditions and increases the degree of safety in unforeseen circumstances. The accommodation of drinking water tanks in the double-bottom space is prohibited.

The drinking water pipeline is made of galvanized steel pipe about 50 mm in diameter for mains and from 13 to 38 mm for branches.

The whole drinking water system must be completely self-contained and the use of its piping, pumps and tanks for other purposes is absolutely prohibited.

A diagram of a drinking water system is shown in Fig. 187. The store tanks, equipped with sounding *4* and vent *3* pipes, are filled through piping *2* which is brought out on the deck. Through inlets *8*, the hand *5* or power-driven *6* pump delivers drinking water to the service tank *7* which has an atmospheric pipe *12* and heating coil *11*. From the service tank *7* water is delivered along mains *9* to the points of consumption *10*. The service tank has an overflow pipe *13* with a valve *14* for returning surplus water back to the store tanks *1*. Connected to the overflow pipe is a branch with the test valve *14* which is brought out at the place where the pumps are installed. The system can be supplied in harbour from shore water mains through hose *15*.

(B) WASHING WATER SYSTEM

Washing water is held in store tanks from where it is pumped to service tanks located on the upper decks and distributed to the points of consumption.

The capacity of the service tanks ranges from 1 to 3 cu m. The tanks are equipped with heaters to avoid freezing and are coated with thermal insulation. The mains are made of steel pipe approximately 50 mm in diameter and the branches are of pipe from 13 to 38 mm in diameter.

The piping diagram of a washing water system in no way differs from that shown in Fig. 187. A water velocity of 0.75 to 1 m per sec is assigned for suction line mains, 1 to 1.2 m per sec for discharge line mains and 1 to 2 m per sec for branches of the discharge line.

(C) SALT (SEA) WATER SYSTEM

Fig. 188 shows an elementary diagram of a salt water system. Sea water is drawn through kingston valve *1* by centrifugal pump *2* or hand pump *3* and is delivered along pipeline *4* to the service tank *5* from where it flows by gravity to the service mains *6* and points of consumption *7*. Service tank *5* is connected with the atmosphere by atmospheric pipe *19* and has an overflow pipe *8* which discharges surplus water overboard. The overflow pipe and the test valve *9* enable the level in the tank to be checked. Through service connection *14*, hose *16* and stop valve *15*, shower *17* or, if necessary, the whole washing water pipeline *18* can be connected to the salt water pipeline. The salt water pipeline can also be supplied from the fire main *10* through reducing valve *12* and stop valve *13*.

Automatic operation of a salt water system can be achieved by the use of pneumatic tanks. A diagram of such a system is shown in Fig. 189.

Water is delivered by hand pump *1* or centrifugal pump *2*, powered by motor *7*, through valve *12* and nonreturn valve *13* into the pneumatic tank *3*. As the water level increases in the tank, the air pressure in it will likewise increase and an air cushion will be formed. At a given pressure in the air cushion, pressure relay *5* switches off the electric motor *7* and stops water delivery to the tank. Then, by the action of the pressure in the cushion, water is delivered along pipeline *8* to the points of consumption *9*. As the water

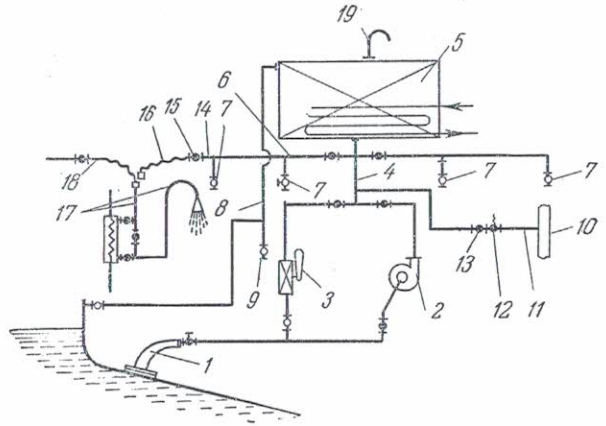


Fig. 188.

is consumed the pressure in the tank drops. When the pressure reaches a definite value the pressure relay *5* switches motor *7* on again through electric circuit *6*, and pump *2* begins to deliver water again to the pneumatic tank.

The centrifugal pump can be cut out of the system by means of disconnecting valves *15*. The tank is equipped with disconnecting *16* and drain *14* valves, and is replenished with air through pipeline *10* and stop valve *11*.

The initial data in designing a pneumatic water supply system are the hourly consumption of sanitary water and the maximum number of times, n_s, the pump is switched on per hour. The latter is usually equal to six.

If the average hourly water consumption is denoted as D, the amount delivered into the tank each time the pump is switched on is

$$D_1 = \frac{D}{n_s} = \frac{D}{6}$$

If V_e cu m is the maximum volume of air at the minimum pressure p_e kg per sq m, and V_f cu m is the minimum volume of air at the

maximum pressure, p_f kg per sq m, then the amount of liquid pumped is

$$V_p = V_e - V_f = D_1 \quad \text{cu m}$$

This equation can be solved for V_e and V_f:

$$V_e = V_f + D_1 = V_f + \frac{D}{6}$$

and

$$V_f = V_e - D_1 = V_e - \frac{D}{6}$$

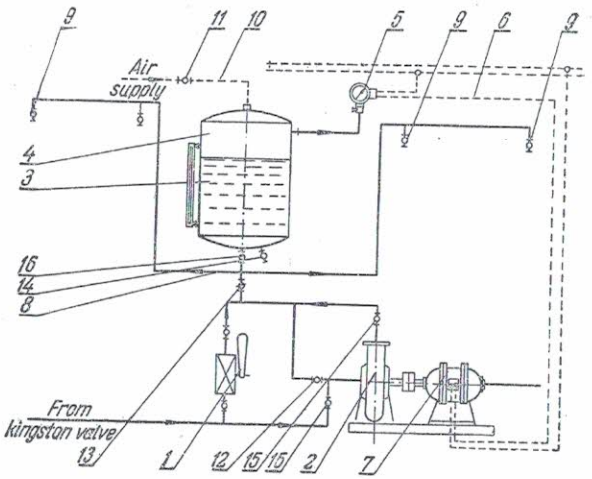


Fig. 189.

The equation of state for the air in the air cushion can be written as

$$V_e p_e = V_f p_f = \left(V_f + \frac{D}{6}\right) p_e = \left(V_e - \frac{D}{6}\right) p_f$$

Therefore the minimum and maximum volumes of the air are

$$V_f = \frac{Dp_e}{6\,(p_f - p_e)} \quad \text{and} \quad V_e = \frac{Dp_f}{6\,(p_f - p_e)}$$

Denoting by V_0 the volume of liquid remaining in the tank at the lowest level, we find that the volume of the pneumatic tank is

$$V_t = V_0 + V_e = V_0 + \frac{Dp_f}{6\,(p_f - p_e)}$$

Such tanks may also be used in the drinking and washing water systems.

(D) SANITARY AND SCUPPER SYSTEMS

The sanitary and scupper systems serve to remove water from the deck and also to dispose of used water from baths, laundries, refreshment bars, galleys, storerooms, etc. Water is drained from the decks through scuppers and their pipes which range from 50 to 100 mm in diameter.

The diagram in Fig. 190 shows how water is removed through scupper pipes *1* from the upper decks and compartment decks. From each deck water runs down to the next lower deck through scupper

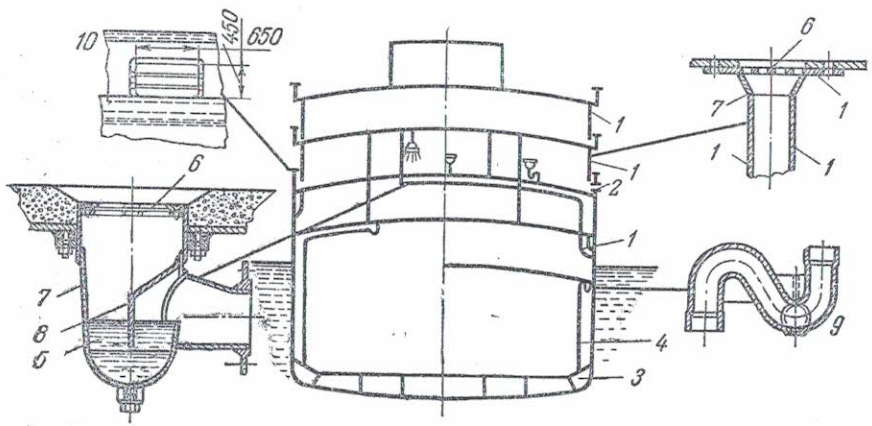


Fig. 190.

pipes until it reaches the last open deck above the load waterline from where it is discharged overboard through deck scuppers *2*. Large amounts of water drain from open decks through freeing ports *10* installed in the bulwarks.

Water is drained from decks located lower than the load waterline through scupper pipes *4* into bilge courses *3* or into dirty water tanks arranged in the double-bottom or side spaces from where it is discharged overboard by pumps.

Scuppers *7* with grates *6*, cowls *8* and sumps *5* avoid clogging of the scupper pipes. S-traps *9* are provided in scupper pipes which drain water from closed compartments to prevent the odour of the sewage spaces from getting into the compartments.

Shipside outlets of scupper pipes serving closed compartments are fitted with swing-check valves to exclude sea water in rough weather.

Sanitary pipelines made of galvanized pipe must be laid with a grade of at least 0.05 to ensure reliable water flow.

(E) SEWAGE PIPELINE

According to the U.S.S.R. Shipping Register the number of water closet bowls on passenger ships is determined from the following data:

(a) one per 50 persons for ships accommodating up to 500 passengers;

(b) $10 + \frac{N-500}{75}$, where N, the number of passengers, is over 500 but less than 1,000;

(c) $10 + \frac{N-1,000}{100}$, where N is over 1,000.

The number of water closets required for the crew is one per 10 to 15 persons. If the crew are accommodated in different sections of the ship, one water closet bowl is to be provided in each section regardless of the number of persons accommodated there.

Water closets are of two types: (1) general use—for unberthed passengers and the crew, and (2) restricted use—for the officers and berthed passengers.

A sewage pipeline is laid of piping at least 100 mm in diameter. It must be as short as possible, have the least possible number of elbows and bends, and a grade of at least 0.05. The sewage pipeline outlet is arranged normal to the plating, about 300 mm above the load waterline and the bend begins not less than 500 mm from the ship's side. A swing-check valve on the outlet opening keeps sea water out during rough weather. If the outlet opening of the sewage pipelines is located below the load waterline or if a water closet may be flooded through the sewage pipeline (even if the outlet is above the load waterline) then a gate valve is mounted in the pipeline at the ship's side before the swing-check valve.

Sewage pipes passing through cargo holds must be protected against damage from cargo handling operations; in refrigerated holds they must be protected against freezing. Wherever possible, sewage piping should not pass through living spaces, provisions stores, storerooms, refreshment rooms, galleys, fresh water tanks, etc.

The outlets of sanitary systems should not be in the vicinity of the holds and salt water inlets. It is not advisable to connect the sanitary and sewage pipelines. The sewage pipeline is washed out by means of the salt water system main.

7-5. Sounding and Vent Pipes

(A) SOUNDING PIPES

The liquid level is checked daily in tanks, collecting wells, and all bilge courses, except those of the engine and boiler rooms, by means of sounding pipes with sounding rods installed over the deepest part of the tank.

A sounding pipe of typical construction, brought out flush with the deck, is illustrated in Fig. 191. An integral or composite pipe *1* passes through the deck plating (Fig. 191*a*). In its upper end it has the plug *5* and, at the lower end, the bottom *2* and openings *3* due to which the liquid in the pipe and in the tank stands at the same level.

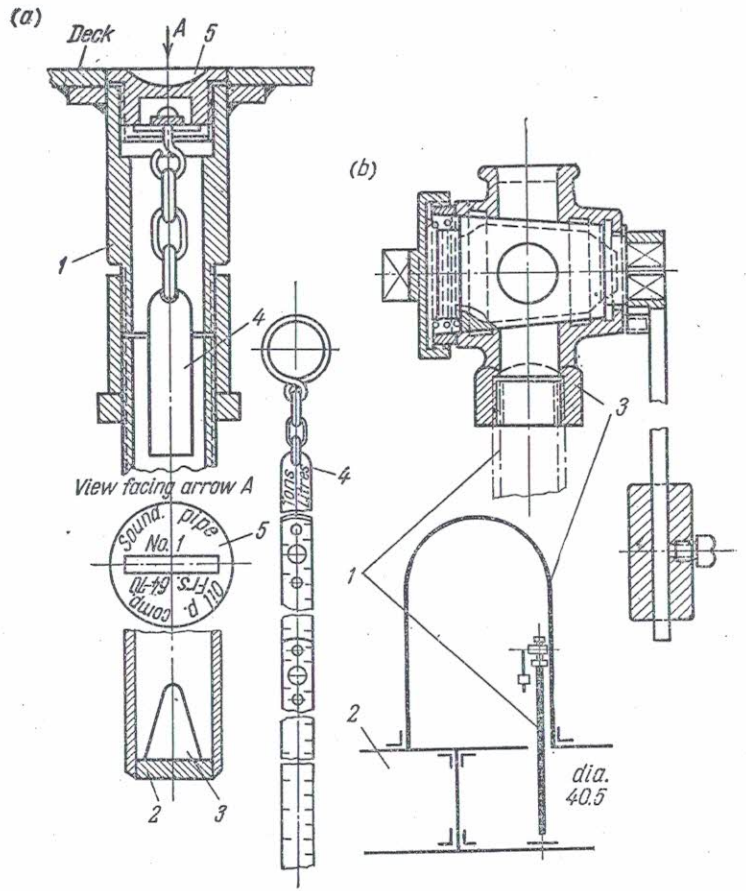


Fig. 191.

The pipe contains a sounding rod *4* on which a scale (1 or 2 m long) is engraved. The scale is graduated in either tons or litres and the amount of liquid in the tank is determined by the length wetted on the scale. When the ship is under way the sounding pipe is closed by an airtight plug on which the pipe number and the name of the space or tank are indicated. If the scale graduations are identical

for a number of sounding pipes, a single portable sounding rod is used for them. As a rule, sounding pipes should be brought out above the maximum load draught of the ship. The sounding pipes *1* of tanks *2* (Fig. 191*b*), located under the deck of the engine and boiler rooms and the propeller shaft tunnel, are brought out in these spaces and equipped with self-closing plug valves *3*. The latter prevent liquid from getting into the engine and boiler rooms if the tanks are overfilled. Galvanized steel pipe 38 mm in diameter is used for sounding purposes. If no bottom is provided at the lower end of the sounding pipe, a steel cover plate is arranged on the ship's bottom under the pipe to avoid damaging the bottom plating with the sounding rod.

On flat bottomed ships or ships that do not have wells in the double-bottom plating, one sounding pipe is installed at each side of the ship. Cocks are provided on the sounding pipes of tanks holding volatile fluids. This enable gases, preventing the liquid from rising to the proper level in the pipe, to be released before making soundings.

(B) VENT PIPES

Vent pipes are installed on all tanks to release the air as the tank is being filled. This ensures complete filling of the tank and avoids pressure increase.

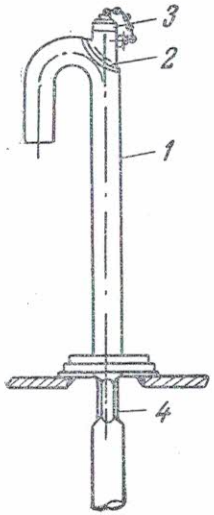


Fig. 192.

Galvanized steel pipe is commonly used for venting. It is laid along the ship's sides and bulkheads and is bent to suit. To avoid choking, the upper end of the vent pipe, brought out on the deck, is bent down through 180° or is equipped with a special cast iron head.

Frequently, vent pipes *1*, installed over the deepest part of a tank, are also used for sounding. In this case, they have a connection *2* and removable plug *3* for inserting the sounding rod (Fig. 192). Air passes from the tank into the pipe and from the pipe into the tank through openings *4*.

Water is prevented from getting into the tanks through the vent pipes, when the decks are washed by waves during a storm, by a ball float valve installed in the bent-over end of the pipe. The floating ball rises and closes the passage in the pipe. According to the U.S.S.R. Shipping Register, vent pipes are to be installed along the ship's sides in each tank and double-bottom space, used for holding liquids, at locations opposite to the filling pipes. If the inner bottom plating is convex, vent pipes should be installed at each side of the centre keel at the highest part of the tank. As a rule, vent pipes from double-bottom tanks under cargo holds, coal bunkers and other spaces that are not always accessible should be brought out on the upper deck or waterproof bulkhead deck. The diameter of vent pipes should be at least 50 mm for double-bottom spaces filled with water and at least 100 mm for spaces filled with liquid fuel.

The total cross-sectional area of vent pipes for water or oil tanks should not be less than (and for deep tanks—1.5 times) that of the piping for filling the tank. In any case, vent pipes of double-bottom tanks for water must have an inside diameter of at least 40 mm or 50 mm if the tanks hold oil products.

The airtightness of the joints of sounding and vent pipes can be checked when the spaces or tanks are tested. This is done by filling the tanks and pipes with water to a height corresponding to the head at which the tank is tested.

The outlets of vent pipes for oil tanks should be equipped with protective devices having screens that can be readily removed for cleaning or repairs. These pipes should be brought out on the upper open deck at a place where the outgoing gases present no fire or health hazard.

7-6. Fire Systems

(A) CAUSES OF FIRES ON VESSELS AND FIRE SYSTEM CLASSIFICATION

Outbreaks of fire on shipboard may be due to a great many different causes.

A fire may be started by such external causes as an excessive increase in temperature in a space adjacent to a bunker and overheating of the bulkhead, the mixing of dry coal stored in the bunker with fresh damp coal, and the spontaneous ignition of coal due to poor ventilation of the bunkers.

The fire hazard on a ship increases with the use of inflammable materials and paint in fitting out the cabins.

The inflammability and combustibility of paint are very high and, since its amount may reach about 100 tons on passenger ships, it is evident that a fire can spread very rapidly once it breaks out.

The most frequent sources of shipboard fires are short circuits due to the poor condition of the electrical wiring, deterioration of the insulation from high temperatures in adjacent spaces and mechanical damage to the insulation during repair operations.

A fire may break out especially readily on an oil tanker when it is loading or discharging cargo.

The air surrounding a tanker during loading or discharging contains a very high percentage of inflammable vapours evolved by the

cargo. The smallest spark of any origin is sufficient to cause an explosion and a fire.

Fire-fighting measures can be classified as prevention, whose aim is to prevent the breaking out of fires, and active measures aimed at extinguishing the fire.

Fire prevention measures are incorporated in the design of a ship, the arrangement and erection of its equipment, etc. These measures are stipulated by the pertinent regulations of the U.S.S.R. Shipping Register.

Active fire-fighting measures are carried out by means of fire-fighting equipment consisting of fire-fighting appliances and fire systems.

Fire-fighting appliances include crowbars, boat hooks, fire axes, felt blankets, sand, hand fire extinguishers, etc., with which a ship is fitted out according to the regulations of the Shipping Register.

Active fire-fighting methods are based upon the following principles:

(1) cooling the burning object with water;

(2) reducing the air content of the compartment or space to 15 per cent, at which concentration combustion ceases;

(3) insulating the burning object from the air by some kind of noncombustible layer.

The purpose of the vessel determines the principle upon which its fire-fighting equipment is based.

It is known, for example, that burning oil products must not be extinguished by a solid stream of water because they rise to the surface of the water and continue to burn. The water may carry away the floating oil and thus spread the fire.

Likewise, burning coal must not be extinguished with water because as the latter contacts the coal it decomposes into hydrogen and oxygen forming an explosive mixture.

It is also impossible to extinguish fires in the engine room with water since oil and liquid fuel may accumulate under the floor plates if the boiler is oil-fired, and in battery rooms since the sea water forms chlorine with the electrolyte of the batteries.

Another reason for refraining from the use of water to extinguish engine room fires is that the water may put the electrical equipment out of order and stop operation of motor-driven fire pumps.

Water may spoil the cargo if used to fight fires in cargo holds. If the hold is completely filled with cargo, preventing direct access to the centre of the fire, a stream of water will prove ineffective.

Fire fighting by cooling is applied on shipboard by means of the fire system which delivers a solid stream of water to extinguish the fire.

The method of lowering the percentage of oxygen in the space or compartment involves the introduction of an inert gas or steam into the compartment, displacing the air until the required volume remains. This reduces the oxygen content to a value that does not maintain combustion.

The principle of insulating the burning object from the oxygen in the air is realized by surrounding the object with an inert foam or a blanket of steam, or by spraying water on it to form a steam blanket.

Fire-fighting systems may be classified as nonchemical and chemical systems according to the method by which the substance is obtained for extinguishing the fire.

Nonchemical systems include water-supplied fire-fighting systems, steam-smothering systems and inert-gas systems.

Chemical systems include foam-extinguishing, carbon-dioxide and mechanically operated foam-air systems. *

(B) WATER-SUPPLIED FIRE-FIGHTING SYSTEMS

Water-supplied systems can be used in all cases to extinguish shipboard fires except when coal or oil products are burning.

The fire system of a ship is centralized and is laid of copper or galvanized steel pipe from 50 to 100 mm in diameter.

The fire main is laid lengthwise along the ship and is equipped with risers spaced not over 20 m from each other.

Hemp hoses, connected to the risers, end in fire nozzles used to direct the stream of water onto the fire.

The fire-extinguishing system of a cargo vessel is shown schematically in Fig. 193. Here branches with risers *2* are connected to the fire main *1*. The U.S.S.R. Shipping Register stipulates that one riser *3* is to be installed directly at each fire pump in the discharge line between the discharge connection of the pump and the disconnecting valve.

If the fire pumps are located in different compartments of passenger ships with well developed superstructures or cargo vessels with a gross tonnage of 4,000 register tons and more, a ring-type fire system must be installed.

Each fire main must be equipped with a device for water supply from a source outside the ship.

The arrangement of the risers should be such as to provide at least two ample streams directed at one spot from hoses not over 20 m long.

* Only the water-supplied systems are dealt with in the present book. Other fire-fighting systems are studied in the course "Safety Engineering and Fire-Fighting Equipment".

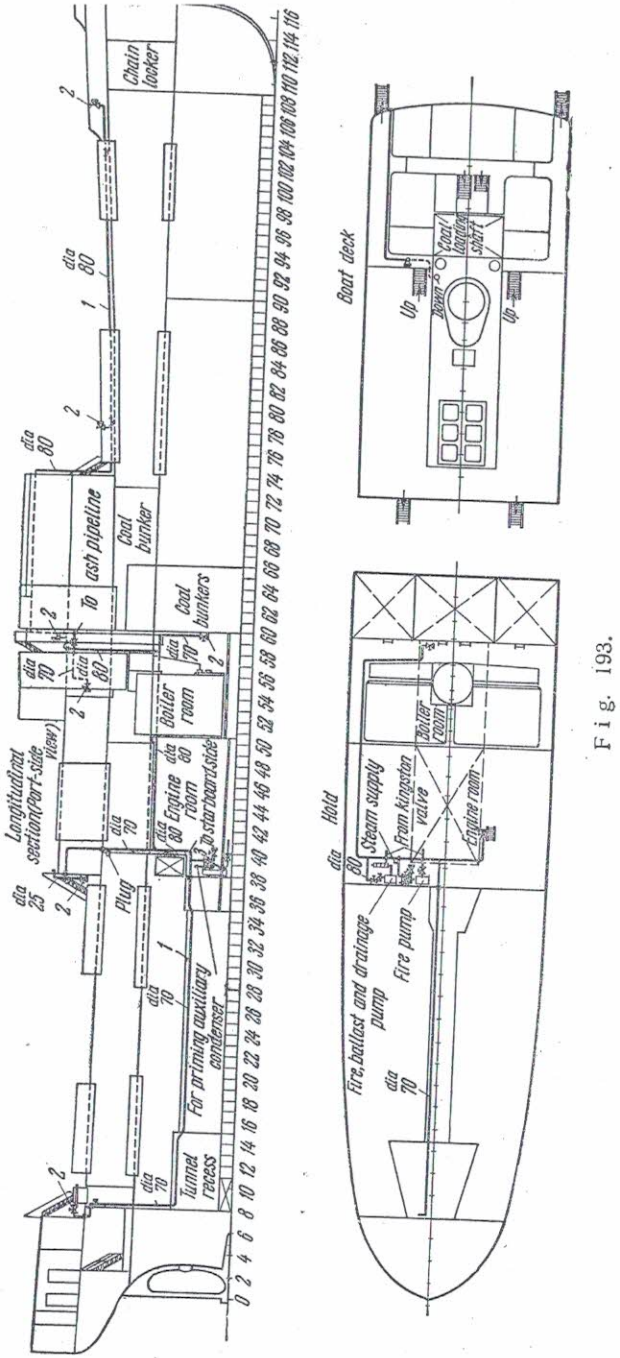


Fig. 193.

At least two fire risers, one at each side of the ship, should be installed in each engine room of motorships and boiler room with boilers fired with liquid fuel on sea-going passenger vessels over 500 register tons and cargo vessels over 1,000 register tons in size.

A fire pipeline should be equipped with drain plugs at the lowest points to drain the water in the winter time.

If the fire pipeline is assembled with flange joints, the jointing material between the flanges must be noncombustible or heat-resistant.

After installation the fire system is to be tested with a hydraulic pressure equal to 1.25 times the conventional pressure in the line. The system is further tested in operation at the working pressure with connected hoses as stipulated by the Shipping Register.

In this test the number of operating risers shall not be less than 15 per cent of the number installed (but not less than three) and the height of the stream, when the nozzle is held vertically, shall be at least 12 m above the deck, highest superstructure or deck house.

In addition to its specific purpose, a fire system is used to hose down the decks, provide actuating fluid for the ash ejectors of the fire rooms and also for the portable ejectors serving to drain water from spaces not accommodated by the drainage system.

Passenger ships carrying over 36 passengers are equipped with a water-sprinkling system that is under a head at all times and operates automatically when fire breaks out in the space or compartment where the sprinkler heads are installed.

Sprinkler heads are installed in cargo holds, engine and boiler rooms, living and service compartments.

In the sprinkler system, the flame is extinguished by a spray formed in the sprinkler head and covering an area with a radius of 3 or 4 m.

The system is under a head at all times and, when the temperature in the compartment increases above a predetermined limit, it automatically begins to operate due to the construction of the sprinkler heads.

The sprinkler head (Fig. 194) consists of the body *1* and threaded ring *2* between which the diaphragm *4* is clamped. The yoke *3* is integral with the ring.

Filler metal brazed around the perimeter of the 12.7-mm hole in the centre of the diaphragm forms a seat *6* for the glass cap *5* serving as a valve disk.

The valve is held tightly in its seat by lock *7* which is soldered in place with an easily fusible solder that melts at a definite temperature. When the solder melts the lock falls apart and valve *5* is opened by the pressure of the water in hole *9*.

The stream of water is sprayed out by deflector *8* to cover a definite area of the compartment.

Sprinkler heads are installed on the basis of one head per 6 to 9 sq m of deck in the compartment, depending upon the fire hazard, and at a distance of 2.5 to 3.5 m from each other. The head of water issuing from the valve should be at least 5 mH_2O.

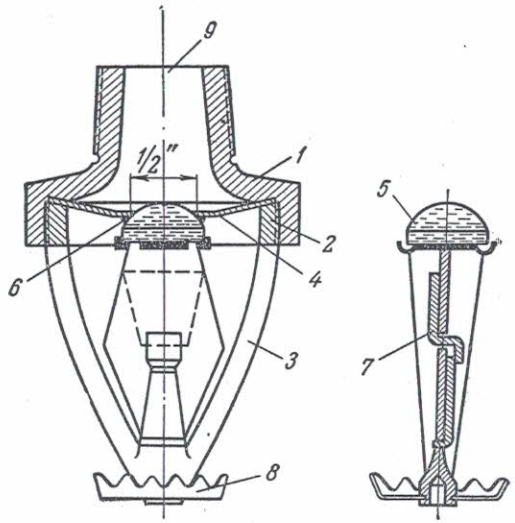


Fig. 194.

7-7. Heating Systems

Shipboard heating systems can be classified as:

(1) Local (stove) heating used on vessels that do not have a mechanical power unit.

(2) Central heating, which may be either steam, hot-water, electric or hot-air, and is used on ships having a mechanical power plant. The hot-air heating system is sometimes combined with the ventilating system of the living and service compartments into a single air-conditioning system.

(A) STEAM HEATING

Steam, used as the heat-carrying agent in this system, is supplied to radiators in the compartments being heated from the ship's boilers or from a source outside the ship (in harbour).

In passing through the radiators the steam gives up its heat through the radiator walls to the air in the room and condenses.

The latent heat of condensation is the main part of the heat utilized for heating purposes.

The lower the steam pressure in the heating system, the higher is the latent heat of condensation, the safer the system is to operate, the less its weight and the less dust will be burned on the surfaces of the radiators. Therefore, the steam pressure never exceeds 2 kg per sq cm. Steam heating systems may be of either the two- or single-main type. A two-main system is illustrated in Fig. 195.

The pressure of the steam supplied in pipeline *1* from the main or auxiliary boilers is higher than the working pressure required for the heating system. The reduction of pressure is effected by throttling the steam in reducing valves *4*.

The steam admitted to the system through pipe *3* is dried in separator *5* installed after the reducing valves *4* to enable the latent heat of condensation to be more fully utilized and to reduce the weight of the heating system.

A relief valve *7* is mounted between reducing valves *4* and separator *5* to avoid pressure increases above the working values.

After the separators the steam passes along pipeline *8* to the distribution boxes *6* and then to the heating radiators *10* from where it emerges as a mixture of condensate and steam and enters the condensate pipeline *12*.

The steam should be completely converted into condensate in the heating system if the latent heat is to be utilized to the greatest extent. This is achieved by installing a steam trap *13* in the condensate pipeline. The trap does not allow steam to escape from the system but condensate drains freely into pipeline *14*.

The condensate of the steam heating system drains to the hot well *15* and from there into the general boiler feed-water system through pipeline *17*.

Provision must be made for a bypass pipeline *18* which enables the system to operate when the steam trap is out of order. A swing-check valve *20* is mounted in pipeline *19* to keep sea water out of the system. Pipeline *2* is for steam supply from a source outside the ship. Stop valves *9* and *11* enable the heating radiators to be disconnected from mains *8* and *12*. Pressure gauges *16* are used to check the live steam pressure in main *8*.

The steam heating pipeline is made of copper or steel pipe and is laid with a grade. Drain cocks are installed at places where condensate may accumulate.

Steam mains are heat-insulated by a coating of asbestos cord sheathed with canvas.

The fittings of the heating system are made of steel in the lines from the boiler to the distribution box; those on the radiators are of bronze.

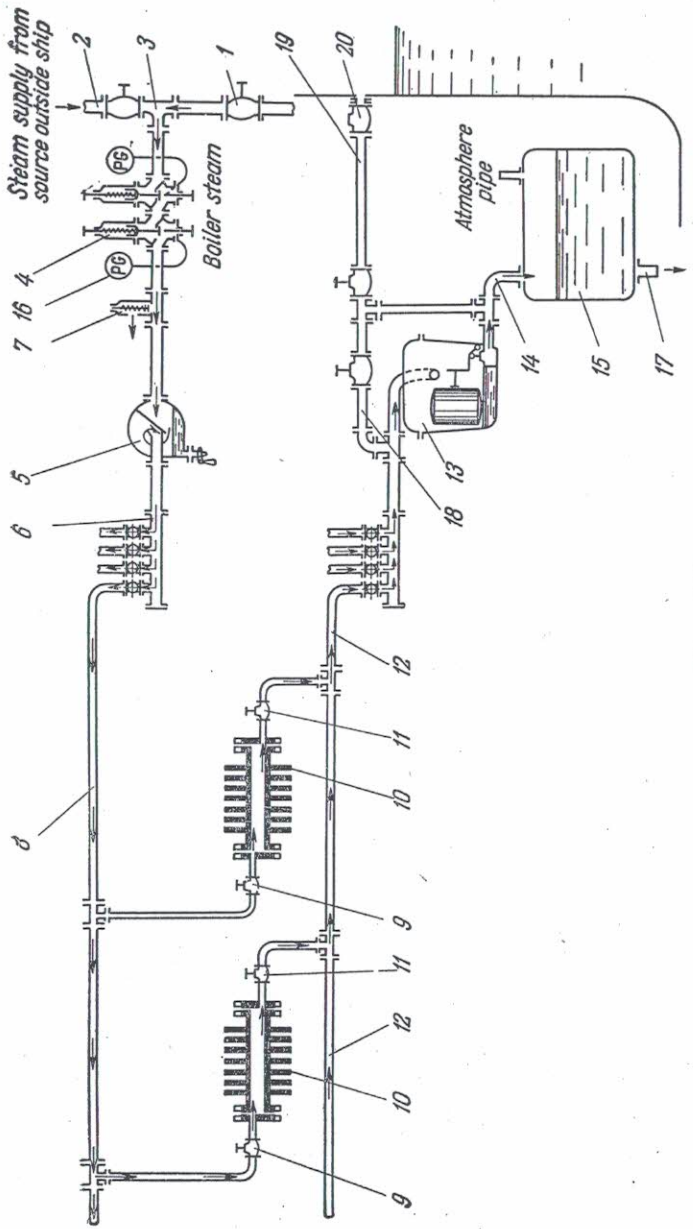


Fig. 195.

In addition to its specific function, a steam heating system is often used as a source of heat for various heating appliances whose pipelines are connected to the heating system.

A diagram of a single-main heating system is shown in Fig. 196. Steam from boiler *1* or from a source outside the ship (along pipe *12*) passes to reducing valve *2*, equipped with a pressure gauge *18*, and then past relief valve *3* and through separator *4* into the distribution box *5*. From the box the steam enters the heating main *6*.

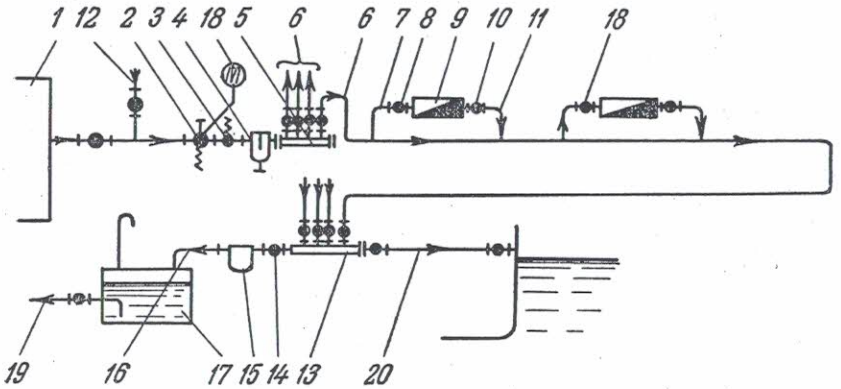


Fig. 196.

The heating radiators *9* are connected to the steam heating main through branches *7*. The amount of steam entering the radiator is regulated by opening valve *8*. The radiator is disconnected from the pipeline by closing valves *8* and *10*.

The steam admitted to the radiator is partly condensed and a mixture of steam and condensate flows down branch *11* into the main and condensate valve chest *13* and further, through valve *14*, into the steam trap *15*. From here it runs through pipe *16* into hot well *17*.

From the hot well the condensate passes along pipeline *19* into a standby tank. Pipeline *20* serves to discharge the spent steam overboard.

Thus, condensate flows along the lower part of the internal surface of the heating main while steam flows above the condensate.

The amount of pipe and fittings required for a single-main heating system is reduced by almost one-half in comparison with the two-main system. This substantially reduces its weight as well.

On the other hand, the heating capacity of the radiators in a single-main system is lower than that in the two-main system because the steam is dampened by the condensate it flows with.

Regardless of the construction of the reducing valve, the steam pressure is reduced by having the steam flow through a regulated passage of reduced area between the valve disk and its seat.

A reducing valve should be designed so that it automatically maintains a constant predetermined pressure to which it is regulated on the low-pressure side, no matter how the steam demand may vary or how the main pressure varies, so long as it does not drop below the reduced pressure required. Reducing valves operate properly when the difference in pressure of the live and throttled steam does not exceed 15 or 16 atm. Regulations of the U.S.S.R. Shipping Register, however, demand two reducing valves be installed in the steam heating main regardless of the pressure drop.

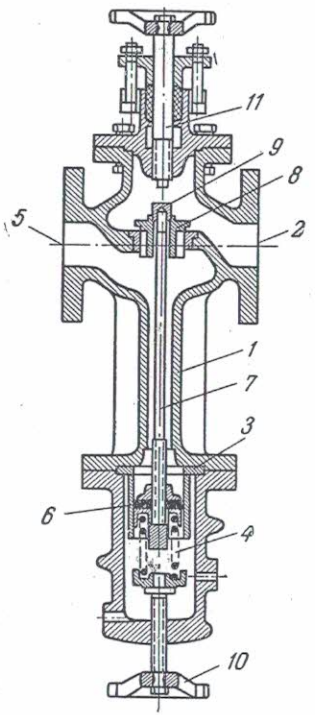


Fig. 197.

Therefore, two reducing valves, combined in a unit with two pressure gauges, are installed in the main even for pressure drops less than the value given above.

As the steam passes through the reducing valve it is dried at the same time that its pressure is reduced.

In a properly operating reducing valve the travel of the valve does not exceed one-fourth of its diameter.

If only one valve is used and the pressure drop is very high, the slot for steam passage through the valve becomes so narrow that automatic pressure regulation of the throttled steam is inaccurate.

Reducing valves may be either of the piston or diaphragm types; they may have either one or two seats.

One of the numerous constructions of reducing valves is illustrated in Fig. 197.

Depending upon the position of the spindle *11* and disk *8*, freely mounted on stem *7*, the valve may either be in the operating position or it may act simply as a shutoff device.

If spindle *11* is screwed down against limiting nut *9*, the disk is held in its seat and out of operation. When spindle *11* is raised, but there is no steam in the passages of body *1*, stem *7* is also raised together with limiting nut *9* while disk *8* rests in its seat.

As soon as live steam is admitted through flange *5* it lifts valve disk *8* up against the limiting nut *9*. When the forces pushing stem *7* upwards are equal to those pushing it downwards, a gap of constant cross-sectional area is established between the disk and its seat. The pressure of the steam will be reduced in passing through this gap and the low-pressure steam will flow through flange *2* to the separator. If the pressure of the steam admitted to the valve drops, the action of spring *4* raises stem *7* and disk *8*. This increases the area of the gap, reducing the velocity of the steam passing through, and the pressure beyond the disk is maintained at the previous value.

An increase in the steam pressure before the valve will increase the pressure of the steam acting on piston *6* in cylinder *3*. This tends to lower the stem *7* and disk *8*, reducing the gap and thereby again maintaining the previous pressure after the valve.

The reducing valve is regulated to the required pressure by turning handwheel *10*.

Separators are based on the principle of suddenly changing or reversing the direction of flow of steam by means of baffle plates, thus throwing out the particles of water, since they are heavier, by centrifugal force so that they fall to the bottom of the separator.

The construction of one type of separator is shown in Fig. 198. Steam, entering the separator through flange *6*, impinges against a baffle having the form of a hollow cylinder *7*, with a pan *4* and ribs *9*, and is directed downwards. After passing around the lower edge of the baffle, the flow of steam is completely reversed by a quick, sharp turn upwards so that it passes up through the inner surfaces and out of the separator through flange *5*. The water separated from the steam runs down pipe *8* and through grate *3* into well *2*. Grate *3* prevents the steam from entraining particles of moisture from well *2* by not permitting it to penetrate deeply under the grate.

Steam traps of steam heating systems must conform to the following requirements: they must operate automatically, prevent the escape of steam, have a space of minimum volume for the condensate, be highly sensitive in operation but simple in construction, not become clogged by dirty condensate, be convenient for disassembly and cleaning, not be susceptible to troubles due to variations in temperatures and pressure and operate normally when the ship rolls or pitches heavily.

According to their principle, steam traps are classified as float, labyrinth and thermal types.

The principle of a float, or bucket, steam trap *13* is clear from its diagram shown in Fig. 195.

The operation of a labyrinth type of steam trap is based on the loss of velocity of steam and its condensation when it passes through

a labyrinth. Such a steam trap is illustrated in Fig. 199. The body *1* contains aluminium washers *2* having locating lugs *6* and clamped in a stack on square-shank bolt *3* by means of nuts. A groove or recess, of variable cross section is provided on the face of each washer. A through hole at one end of the recess connects the recess of one washer with that of the adjacent washer. Thus, the recesses constitute a sinuous passage for the steam and condensate.

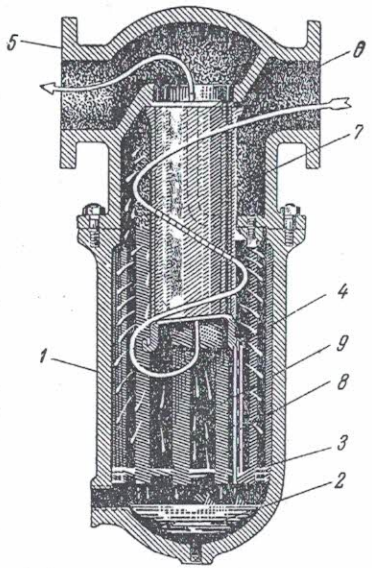


Fig. 198.

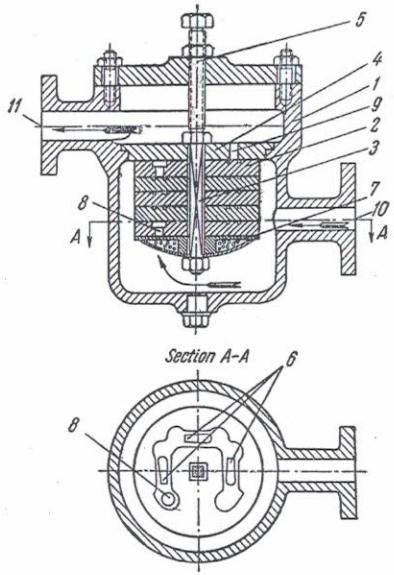


Fig. 199.

Cover *4* and screw *5* hold the stack of washers firmly in body *1*. The trap is connected by flange *10* to the condensate pipeline. The condensate fills the space in the body and flows through screen *7* and hole *8* in the lower washer to enter the labyrinth. From the labyrinth it flows through hole *9* and flange *11* to the hot well. When steam enters the trap it expands repeatedly in the labyrinth, due to the variable cross section, loses its pressure and velocity and condenses.

The thermal type of steam trap is shown in Fig. 200. Flexible bellows *2* is brazed to cover *6* of the trap. It has a needle valve *3*. The bellows contains a volatile liquid and is filled through the hole closed by plug *4*. When steam enters the trap from pipeline *7*, the liquid in the bellows vapourizes and the vapour pressure stretches the bellows, thereby closing the needle valve *3*.

After the steam condenses and the temperature of the medium surrounding the bellows drops, the vapour in the bellows also condenses. As a result, the pressure in the bellows drops, the bellows contacts and raises needle valve *3*. The condensate filling the space in body *1* then passes through the valve seat and connection *5* to the hot well. Such traps should be installed at each radiator for more effective utilization of the latent heat.

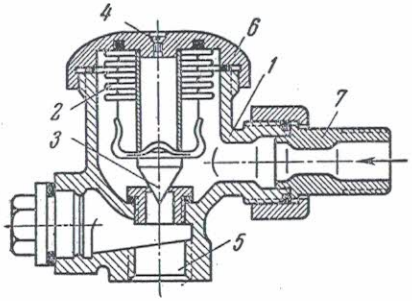


Fig. 200.

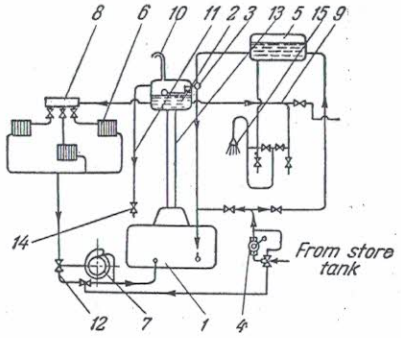


Fig. 201.

(B) HOT-WATER HEATING

A diagram of a hot-water heating system is shown in Fig. 201. Water is heated to 80 or 90° C in boiler *1* from where it flows through riser *13* to expansion tank *2* which is equipped with signal pipe *11* and valve *14*.

Expansion tank *2* receives a part of the water in the system when it expands as the system is being started up and allows steam to escape freely through atmosphere pipe *10* thus avoiding steam pockets in the system.

From the expansion tank water flows to valve chest *8* and radiators *6* and then to pump *7* which delivers it back to the boiler *1*.

The difference in the temperature of the water heated in the boiler and of that cooled in the radiators establishes a natural circulation when the pump is switched off. This is made use of by the provision of bypass pipeline *12*.

Losses of water in. the system are made up by pump *4* which delivers water from storage tanks to tank *5*. From tank *5* water flows into the boiler through valve *3* which is operated automatically by a float, depending upon the water level in the expansion tank.

Water required for showers *15* and other domestic needs is tapped from the expansion tank by pipeline *9*.

Compared to steam, the advantages of hot-water heating are:
(1) it is more sanitary;
(2) it continues to operate longer after the water is not heated any more;
(3) it gives longer service;
(4) it is silent in operation;
(5) it operates without pressure;
(6) it is simpler in construction.

Disadvantages include:
(1) the larger weight of the system;
(2) higher initial cost of installation;
(3) danger of freezing in the pipes;
(4) slower heating of compartments due to the lower temperature of the heating agent and its low rate of flow (not over 0.3 m per sec);
(5) difficulty in achieving uniform heating of the radiators.

The surface of radiators for hot-water heating must be 1.4 times greater than that of steam radiators to obtain the same temperature conditions in a compartment.

Water from the cooling system of internal combustion engines can be used as the heating agent in a hot-water heating system.

From the point of water distribution, the piping must be laid with a grade of least 5 to 10 mm per metre to avoid standing of water in the pipes and its consequent freezing.

The parts of the heating pipeline are to be subjected to hydraulic strength tests at a pressure of 4 kg per sq cm, and the assembled system—at a pressure of 1.5 kg per sq cm.

(C) DESIGN OF STEAM AND HOT-WATER HEATING SYSTEMS

Steel or cast iron finned pipe and radiators are used as heating appliances of shipboard steam heating systems.

A disadvantage of finned pipe is that dirt quickly fills the spaces between the fins. In operation of the system this dirt burns on the surfaces and emits an unpleasant odour.

The main drawback of radiators is their excessive weight. The weight, for example, per square metre of heating surface is 2.5 times larger than that of a finned pipe with the same heating surface. Radiators, however, are more sanitary than other appliances.

The amount of heat dissipated in each shipboard compartment is

$$Q_c = F_c (t_i - t_o) k_c$$

where F_c = area of the surface surrounding the compartment, sq m
t_i = temperature of the air in the compartment, °C
t_o = temperature of the outside air, °C

k_c = coefficient of heat transmission of the surfaces surrounding the compartment, kcal per hr per sq m per deg C.

The surface of any given compartment or space is determined from the drawings of the ship. The required air temperatures inside various compartments are given in Table 39.

In heating calculations, the temperature of the outside air in winter is taken as 15° C below zero for the Baltic, Black and Caspian Seas and 35° C below zero for the North Sea and Pacific Ocean.

The heat required in heating the ventilating air supplied to the compartment is

$$Q_v = V_c n \gamma_a c_a (t_i - t_o) \quad \text{kcal per hour}$$

where V_c = volume of the compartment, cu m
n = number of air renewals per hour
$\gamma_a = 1.213$ = density of air at 18° C, kg per cu m
$c_a = 0.241$ = heat capacity of air, kcal per kg per deg C.

Therefore, the amount of heat required to maintain a definite temperature in a compartment is

$$Q_v = Q_c + Q_v \text{ kcal per hour}$$

The required surface, F_r, of the radiators for a given compartment is found from the formula:

$$F_r = \frac{Q_r}{k_r (t_m - t_i)} \quad \text{sq m}$$

where t_m = mean temperature of the heating agent, ° C
k_r = coefficient of heat transmission of the radiator or other heating appliances (Table 68), kcal per hr per sq m per deg C.

Table 68

Type of heating appliance	k_r, kcal/hr/m²/deg C
Plain radiators:	
for steam	8 to 9
for hot water	5.7 to 7.3
Finned cast iron piping:	
for steam	6.5
for hot water	4 to 5.5
Finned steel piping for steam	8
Plain steel piping for steam, laid horizontally, d=30 to 150 mm	11 to 12
Ditto, but copper for steam	15

If the radiator is protected by guards or hoods, the calculated surface must be increased by 10 to 20 per cent.

The total amount of heat emitted by all the radiators is

$$Q = \Sigma Q_r \quad \text{kcal per hour}$$

The steam consumption of a steam heating system is

$$D_s = 1.1 \frac{Q}{\lambda_s} \quad \text{kg per hour}$$

where $\lambda_s = 500$ to 520 = effectively utilized heat of the steam in the radiators, kcal per kg

1.1 = coefficient taking into account steam losses through leaks and in the steam trap.

The required heating surface of the radiators or other heating appliances can also be determined from the formula:

$$F_r = \frac{V_c}{k_{ra}} \quad \text{sq m}$$

Values of the coefficient k_{ra} are: 30 for internal cabins, 10 for external upper deck cabins and 20 for cabins of other decks.

A tentative estimate of the steam consumption for steam heating and various domestic needs is given by the formula:

$$D_s = 0.7 V_c^{0.5} (17 - t_o) + 0.2 n_{cp} \quad \text{kg per hour}$$

where n_{cp} = number of persons on board ship (crew and passengers).

(D) HOT-AIR HEATING

A hot-air heating system usually serves as the shipboard ventilating system as well. Consequently, fresh air is used as the heating agent.

The combination of ventilation and heating in a single system permits the temperature of the air in any compartment to be varied in a wide range.

The air is heated in heaters and cooled in coolers, but sometimes the same heat-exchange device, called a thermo-tank, is used for both purposes.

Evidently, the amount of heat carried into the compartment by the air should be equal to the amount lost there. Thus

$$Q_c = G_h c_a (t_d - t_a) \quad \text{kcal per hour}$$

where G_h = required amount of air, kg

t_d = temperature of the air delivered to the compartment, °C

t_a = normal air temperature in the compartment, °C

c_a = heat capacity of air, kcal per kg per deg C.

The volume of air required for heating is

$$V_h = \frac{G_h}{\gamma_t} = \frac{G_h c_a (t_d - t_a)}{\gamma_t} \quad \text{cu m}$$

where γ_t = density of air at the given temperature, in kg per cu m.

In cases where the hot-air heating and the ventilating systems are combined, the volume of air supplied to a compartment must meet the requirements for both systems.

If the amount of air required for ventilation turns out to be greater than that required for heating, then its temperature should be

$$t_d = t_a + \frac{Q_c}{c_a G_v} \leqslant 40° \text{C}$$

where G_v = amount of air required for ventilation, kg.

(E) ELECTRIC HEATING

Electric heating is based upon the conversion of electrical energy into heat energy which is transmitted to the air in the compartment by the heating appliances.

An electric heating system consists of:

(1) switchboard panels with the necessary switches, instruments and protective devices;

(2) machinery for producing electrical energy, transmission lines and apparatus for turning the heaters in the compartments on and off and for regulating them;

(3) resistance heating units of appropriate design called heaters.

Electric heaters may be either stationary, wall-mounted or portable; they may be of either open or closed design.

Electric heating systems have the following advantages over other systems:

(1) their initial cost is lower;

(2) dust is not burned on the heaters;

(3) the whole equipment has a low weight;

(4) they are exceptionally convenient in operation.

The chief drawback of such systems is the high cost of operation, being twice that of a steam heating system.

In designing heaters it is usually necessary to determine the required length of the resistance wire in the heating element.

Denoting the amount of heat losses in a compartment as Q kcal per hour, the current flowing in the heating elements is

$$i = \frac{Q}{0.864 v} \quad \text{A}$$

where v = mains voltage, volts.

The resistance of the heating element is

$$R = \frac{v}{i} \quad \text{ohms}$$

The length of the resistance wire is

$$l = \frac{RA}{\rho}$$

where A = cross-sectional area of the wire
ρ = specific resistance of the wire, ohms.

The power of the heaters in a d-c system is

$$P = \frac{Q}{0.86} \text{ W}$$

The following empirical formula can be used for tentative power requirement calculations of electric heaters:

$$P = 0.53(5.4F_c + 10.8F_p + 0.4V_c)(t_i - t_o) \quad \text{W}$$

where F_c = area of the deck, sides and bulkheads in the compartment, sq m
F_p = area of port lights and skylights, sq m
V_c = volume of the compartment, cu m
t_i = required air temperature in the compartment, ° C
t_o = temperature of the outside air, ° C.

In the operation of an electric heating system the heaters operate at their rated power only for the first 2 or 3 hours until the compartment is heated up to the required temperature.

7-8. Ventilation

(A) PURPOSE AND PRINCIPLES OF VENTILATION

The function of a ventilation system is to maintain the required chemical composition and humidity of the air in the shipboard compartments.

The composition of the air in a compartment with people is changed by the carbon dioxide, moisture, heat and, with them, the products of protein decomposition produced by the occupants.

The percentage of the products of the physiological decomposition of protein substances in the air of a compartment increases in proportion with carbon dioxide content. If the air contains over 0.15 per cent carbon dioxide, the products of protein decomposition have a harmful effect on the human organism. The chemical composition of the air we breathe is: 20.96 per cent oxygen, 78 per cent nitrogen, 0.03 per cent carbon dioxide and 1.01 per cent inert gases.

Exhaled air contains approximately 16.4 per cent oxygen, 73.6 per cent nitrogen, 4.6 per cent carbon dioxide and 1 per cent inert gases.

Thus, the amount of oxygen in the exhaled air is reduced by 1.5 times while the amount of carbon dioxide increases 150 times.

A person can endure a carbon dioxide content of the air up to 1 per cent for a comparatively long time, up to 4 or 5 per cent for a half-hour or an hour without grave consequences, and from 6 to 8 per cent for the same period but with a serious aftereffect.

Normal air for breathing purposes should not contain more than 0.1 per cent carbon dioxide.

A person develops from 75 to 150 cal of heat per hour depending upon whether he is active or at rest.

The heat developed by a person is dissipated by contact of the skin with the air in the compartment, by radiation, by the evaporation of moisture from the surface of the body and by the carrying away of heat with the exhaled air.

The amount of heat developed by a person depends only to a small extent upon the temperature and humidity of the air. On the other hand these factors vary the percentage of heat dissipated in the different ways indicated above.

Thus, at an air temperature of 37° C radiation and contact are ineffective since the body has this same temperature.

In this case, the developed heat can be removed only by evaporation of exuded moisture and in the exhaled air, if the air in the compartment is sufficiently dry.

This is the explanation of the well-known fact that the human body can withstand temperatures above 37° C if the ambient air is dry enough.

The foregoing leads to the conclusion that the amount of air required for the ventilation of a given compartment should be determined on the basis of:

(1) maximum permissible air temperature in the compartment;
(2) maximum permissible air humidity in the compartment;
(3) maximum permissible percentage of carbon dioxide in the air of the compartment.

The condition of the air in engine and boiler rooms depends to a considerable degree upon the heat, gases and vapours evolved by the power plants.

The cargo carried in the holds requires a definite condition of the air for its preservation. This is achieved by disposing of the emissions of the cargo and by replacing the vitiated air with fresh air.

Fuel bunkers also require ventilation to remove the vapours evolved by oil products or the gases generated by coal. The temperature of coal is reduced by ventilation to prevent self-ignition.

Thus, shipboard ventilation is divided into:

(1) ventilation of the living and service compartments with the purpose of exhausting vitiated air, unsuitable for breathing, and replacing it with fresh air;

(2) ventilation of the engine and boiler rooms designed for lowering the temperature and maintaining the required composition of the air in them;

(3) ventilation of the cargo holds with the purpose of maintaining the composition and humidity of the air required to preserve the quality of the cargo;

(4) ventilation of the bunkers to remove gases evolved by the stores of fuel.

Ventilation systems may be either natural or mechanical. Natural ventilation is a system in which air renewal is the result of natural forces, for example, heat flow, wind, etc.

In a mechanical system of ventilation fans are employed to effect air renewal.

Conditions favourable for natural ventilation are created by leakages in the compartment and the temperature difference between the inside and outside air.

If air can leak into the compartment at the bottom and leak out at the top and if the temperature in the compartment is higher than that of the outside air, i.e., $t_i > t_o$, an upward flow of air will be produced.

The air in the compartment is lighter than the outside air and therefore it rises. This increases the pressure at the top and reduces it at the bottom of the compartment.

As a result, air will be discharged from the compartment through the upper leakages or openings and outside air will be drawn in through the lower openings. Since the pressure of the air is higher at the top of the compartment and lower at the bottom than that of the outside air, at a certain height the inside and outside pressures must be equal.

The cross section of equal inside and outside pressures is called the neutral plane of the compartment. If the leakages at the top and bottom are the same and located symmetrically, the neutral plane will pass through the middle of the compartment.

Leakages in ship compartments, such as port lights, doors, hatches and vent holes, are always asymmetrically located. Therefore, warm air from the compartment passes out through openings above the neutral plane and fresh cold air enters the compartment below this plane.

Vertical ventilation ducts are arranged above compartments to reinforce the natural ventilation. They utilize the difference in temperature t_i of the air in the duct and the temperature t_o outside

of it. The pressure of a column of air, H m high, acting from above in the duct on 1 sq m of its cross-sectional area is

$$p_i = p_e + \gamma_i H \quad \text{kg per sq m}$$

where p_e = pressure of the external air, kg sq m

γ_i = density of air at temperature t_i, kg per cu m.

The pressure of a column of outside air acting from below on the same cross-sectional area is

$$p_o = p_e + \gamma_o H \quad \text{kg per sq m}$$

The difference in pressure of the external air and the air of the compartment in the duct produces the necessary conditions for natural ventilation and is

$$\Delta p = p_o - p_i = (p_e + \gamma_o H) - (p_e + \gamma_i H) = H(\gamma_o - \gamma_i) \quad \text{kg per sq m}$$

The efficiency of natural ventilation is usually increased by utilizing wind pressure, employing vent heads and deflectors for this purpose.

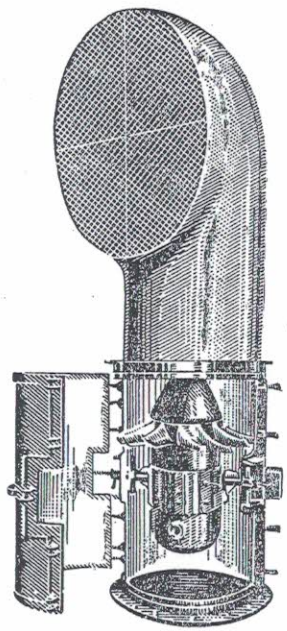


Fig. 202.

Mechanical ventilation can be designed along centralized lines in which the ventilating action is effected by fans connected to ventilation mains that supply the various compartments through branch ducts. Self-contained mechanical ventilation is also available. In this case, each deflector (funnel) has its own fan (Fig. 202).

Mechanical ventilation may be plenum, exhaust or a combination of the two (plenum-exhaust ventilation). In plenum ventilation fresh air is supplied to the compartment by a fan, and vitiated air is removed through leakage due to the increased pressure in the compartment produced by the supply fan.

In exhaust ventilation, the vitiated air is removed by the fan, and fresh air enters the compartment by leakage due to the drop in pressure produced by the exhaust fan.

In plenum-exhaust ventilation the air is both supplied to and exhausted from the compartment by fans.

As a rule, plenum ventilation is installed in spaces where a supply of fresh air is required or it is necessary to raise the pressure to prevent the infiltration of vitiated air from adjacent compartments.

Exhaust ventilation is employed for compartments that contain constant sources of air contamination or those to which natural access of air is difficult.

In tentative calculations, the required amount of air is determined on the basis of the number of air renewals per hour as established by practice for various compartments and indicated in Table 42.

The capacity Q of deflectors and vent headers for natural ventilation is calculated for a mean air flow $v = 2$ to 4 m per sec according to the following formulas:

for plenum ventilation

$$Q = 3{,}600\ vf\frac{\gamma_o}{\gamma_i} \quad \text{cu m per hour}$$

and for exhaust ventilation

$$Q = 3{,}600\ vf \quad \text{cu m per hour}$$

where γ_i and γ_o = density of air in the compartment and of fresh air, respectively, kg per cu m

f = active area of the deflector ($f = 0.785\ d^2$ for a round pipe with a diameter d and $f = \frac{3.14(ab)^2}{(a+b)^2}$ for a rectangular pipe with sides a and b), sq m.

Chapter 8

SPECIAL SYSTEMS OF ICE-BREAKERS AND OIL TANKERS

8-1. Heeling and Trimming Systems of Ice-Breakers

Heeling tanks are arranged in the ship's hull so that they do not affect the trim of the ship, develop the maximum heeling moment and reduce the restoring moment when ballast is taken in.

Fig. 203 shows a diagram of the heeling system of an ice-breaker with side heeling tanks *1*, having a capacity of 200 cu m each. The ballast is transferred from tank to tank by the reversible pump *2* having an output of 135 tons per min. Thus, the whole water ballast

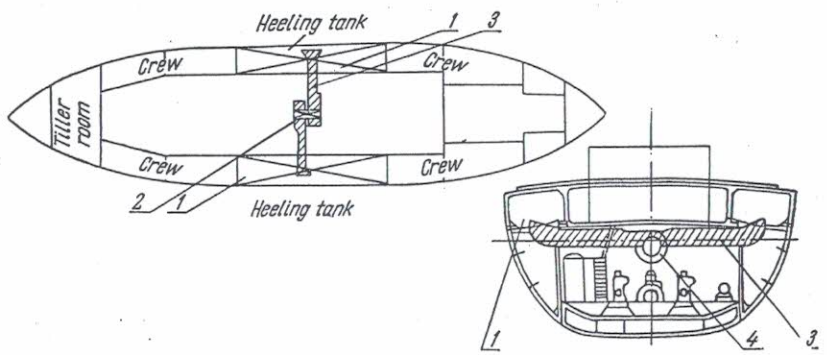


Fig. 203.

can be pumped over from side to side in 1.5 minutes along the square pipe *3* (cross-sectional area 1 sq m) to produce a heel of 8°.

The water ballast used to obtain the required trim is held, for convenience in manoeuvring, in several trimming tanks located in the fore and after ends of the ship. The total capacity of these tanks is from 6 to 7 per cent of the ice-breaker's displacement. A line diagram of the trimming system of an ice-breaker is shown in Fig. 204. It has a centrifugal pump with an output of 1,200 cu m per hour, the capacities of the fore and after tanks being 132 and 135 cu m, respectively. It takes about 7 minutes to transfer the

ballast from one tank to the other. Various positions of valves *1*, *2* and *3* and register (regulating valve) *R* enable pump *5* to deliver water, entering through kingston valve *K*, to either the fore or after trimming tanks, or to transfer water from the after to the

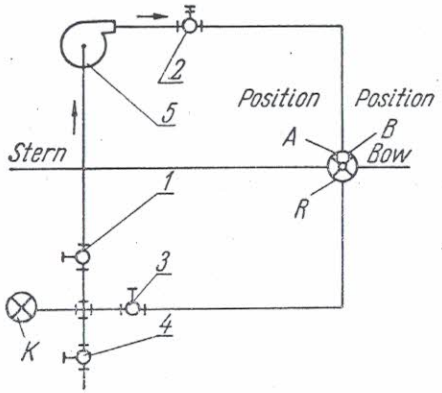


Fig. 204.

fore tank or vice versa. By means of valve *4* the trimming tanks can be connected to the ballast pump which serves to prime the trimming pipeline and pump before starting the latter.

8-2. Special Systems of Tankers

The cargo system of a tanker must provide for: (1) loading and discharging as rapidly as possible; (2) emptying the tanks as completely as possible and (3) simplicity in construction and operation. Moreover, the cargo system must be safe against fires.

A liquid cargo is loaded by shore and shipboard pumping plants and discharged by shipboard facilities through the same cargo pipeline. The diameter of this pipeline must be such as to ensure the flow of oil products at a velocity from 2 to 2.5 m per sec. The cross section of the inlet branches in the tanks should allow the continuous discharge of two tanks simultaneously. The suction main is laid on each side of the centre line bulkhead and usually above the bottom framing at a height of 1,200 to 1,500 mm from the bottom. This main may pass through the transverse framing to the discharge pumps only when very heavy oil products are to be handled.

The port and starboard mains are cross-connected in the end tanks and pump room by a transverse pipeline with disconnecting valves.

Expansion joints, installed to relieve stresses due to thermal deformation of the piping, are of the stuffing-box type with a hemp or asbestos packing. The pipeline is mounted on at least three cast iron brackets in each tank to eliminate deformation in operation. Gate valves are used exclusively as shutoff fittings in the cargo pipeline and they are located as close as possible to the inlet connection.

Since the shutoff fittings are controlled from the deck, the most critical parts are the stems of the gate valves. Special attention must be paid to the strength and design of these stems.

In addition to the main tanks, summer tanks are installed on the filling tanks. They are located in the 'tween-deck space and carry the lighter grades of cargo. Summer tanks were thus named because previously they served to carry the overflow from the main tanks due to expansion of the liquid cargo when navigating under summer conditions. At the present time each cargo tank is provided with a trunk for this purpose.

If the summer tanks are to carry the same grade of cargo as the main tanks, they are discharged through valves cut into their bottoms and connected by piping with the main cargo pipeline. They are loaded through these same gate valves to the level of the liquid in the expansion trunks of the main tanks. Then the drain gate valve of the summer tanks is closed and further loading is carried out through their hatch covers. If different cargoes are to be carried, a separate pipeline with an independent pump is installed for the summer tanks. The inlet connections of the cargo system terminate 100 to 150 mm short of the bottom of the tanks and therefore the tanker cannot be completely emptied.

Tanks with a liquid cargo are drained by a special evacuating pipeline having an independent pump whose output is 25 per cent of that of the main cargo pump. This enables the inlet branches to be arranged 15 to 20 mm above the bottom of the tank so that the cargo can be completely discharged.

Since at low temperatures the viscosity of oil products increases to a degree that makes flow through piping impossible, tankers must be equipped with a cargo-heating system.

This heating system consists of tubular heaters, about 50 mm in diameter, into which steam is admitted at a pressure from 2 to 7 kg per sq cm. The heating system is to be hydraulically tested at twice the working pressure of the heating steam.

The heating system is equipped with steam traps to effect complete condensation of the steam, and the piping is laid with a grade to obtain better drainage of the condensate. The suction connections of liquid fuel tanks are fitted with heated strainers.

Heat carried to the coils of the cargo-heating system is expended (1) for heating the cargo, (2) as losses through decks and sides that are in contact with the outside air, and (3) as losses through the sides and bottom that are in contact with the water.

The amount of heat, Q_1, required to heat the cargo is

$$Q_1 = 1{,}000\, V\gamma\, [mc_1 + c_2 (t_2 - t_1)] \quad \text{kcal}$$

where V = volume of the hold, cu m
γ = density of the oil product, kg per cu m
t_1 and t_2 = its initial and final temperatures, °C
c_1 = melting heat of solidified components of the cargo that are found only in paraffinaceous masuts (for paraffin $c_1 \approx 50$ kcal per kg)
m = proportion of paraffinaceous components in the cargo (for paraffinaceous masuts $m \approx 0.05$)
c_2 = heat capacity of the oil product, kcal per kg per deg C (for masut $c_2 \approx 0.5$).

Heat transmission through the deck and sides to the outside air is

$$Q_2 = k_1 F_1 \left(\frac{t_1 + t_2}{2} - t_3 \right) \quad \text{kcal per hour}$$

where $k_1 \approx 7.5$ to 10 = coefficient of heat transmission from the cargo to the outside air, kcal per sq m per hr per deg C
F_1 = surface of the sides and deck that are in contact with the outside air, sq m
t_3 = temperature of the outside air, ° C.

The heat transmission through the sides and bottom to the sea water is

$$Q_3 = k_2 F_2 \left(\frac{t_1 + t_2}{2} - t_4 \right) \quad \text{kcal per hour}$$

where $k_2 \approx 34$ to 49 = coefficient of heat transmission from the oil product to the sea water, kcal per sq m per hr per deg C
F_2 = surface of contact of the sides and bottom with the sea water, sq m
t_4 = temperature of the sea water, °C.

The amount of heat required to heat the cargo to the required temperature in τ hours is

$$Q' = Q_1 + Q_2 \tau + Q_3 \tau \quad \text{kcal}$$

Therefore, the hourly heat consumption is

$$Q = \frac{Q'}{\tau} = \frac{Q_1}{\tau} + Q_2 + Q_3 \quad \text{kcal per hour}$$

The heating surface of the heater is found from the formula:

$$Q = F k_3 \left(t_5 - \frac{t_1 + t_2}{2} \right) \quad \text{kcal per hour}$$

where $k_3 \approx 50$ = mean coefficient of heat transmission from the steam to the cargo, kcal per sq m per hr per deg C
t_5 = mean temperature of the steam in the heating coil, °C
$\frac{t_1 + t_2}{2}$ = mean temperature of the cargo during the time it is heated, °C.

Then the heating surface of the coil is

$$F = \frac{Q}{k_3 \left(t_5 - \frac{t_1 + t_2}{2} \right)} \quad \text{sq m}$$

The length of pipe with an outside diameter d needed for the coils is

$$l = \frac{F}{\pi d} \quad \text{m}$$

The steam consumption required to heat the cargo is

$$D = \frac{Q}{i'} \quad \text{kg per hour}$$

where $i' = 500$ to 550 = effectively utilized heat of the steam, kcal per kg.

The mean rated velocity of the steam in the piping should not exceed 10 to 30 m per sec and is determined from the formula:

$$w = \frac{4D}{11{,}300 \gamma_s d_1^2 z} \quad \text{m per sec}$$

where γ_s = density of the steam, kg per cu m
z = number of coils connected in parallel
d_1 = inside diameter of the pipe, m.

Calculations are made for the coldest period of the year with the cargo being heated to the required temperature in several days. In more precise calculations, the logarithmic difference of the temperatures should be used in the formulas in place of the arithmetic difference, and the coefficients of heat transmission should be determined from the pertinent formulas.

As a rough estimate of the heating surface of the coils required per ton of cargo, the following values can be used: 0.03 sq m for medium-viscosity crude oil, 0.046 sq m for high-viscosity crude oil and 0.09 sq m for extra-high-viscosity crude oil.

The functions of the gas-discharge system are to dispose of the vapours evolved by the liquid cargo, remove air from the tanks when they are being loaded and to prevent the creation of a vacuum when the tanks are discharged. This system is designed either as separate gas-discharge pipes for each tank (self-contained system), or the gas-discharge pipes of a group of holds are connected to a common main (group system). The gas-discharge pipes are installed at the highest point of the hold or on the covers of the trunking (trunking is provided to reduce the evaporating surface of the cargo). These pipes range from 75 to 100 mm in diameter.

The outlet holes of gas-discharge pipes should not be located nearer than one metre, in both the horizontal and vertical directions, from the nearest doors, port lights or other similar openings. The height of the pipes, as measured from the point where they are connected to the tank, should be at least 2 m and their outlet ends should be equipped with fire screens.

Fire screens are mounted on the upper ends of separate gas-discharge pipes. In the group system, the fire screens are installed at the outlet holes and also at the places where pipes enter the tanks. During a fire, the former prevent the flame from getting into or out of a hold, while the latter prevent the penetration of flame from hold to hold.

In carrying first-category oil products, the cargo loss, due to evaporation and free exit of the vapours through the gas-discharge system, can be reduced by installing breather valves in this system. A breather valve consists of a pressure-actuated valve which opens when the pressure in the hold reaches 50 to 60 mm H_2O, and a vacuum valve which opens when a vacuum of 100 mm H_2O is created in the hold. The first serves to release air in loading the cargo and gases and vapours in carrying the cargo. The second prevents a vacuum from being created when the cargo is discharged.

The purpose of the deck-flooding system is to reduce losses of a liquid cargo caused by intensive evaporation in hot weather.

This system consists of a pump, gravity tank and pipeline laid along the deck. The latter is flooded through small sprinkling holes in the piping. The holes have a diameter of 3 or 4 mm and are spaced from 40 to 60 mm apart along the pipe. The water has a pressure of about 0.5 kg per sq cm at the holes. The discharge through the deck-flooding system should be calculated so that the water covers the deck in a thin layer. The maximum effect is achieved if the water evaporates as it runs off the deck.

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TO THE READER

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